Steam, Air, and Gas Power

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FIFTH EDITION

JOHN WILEY & SONS, INC., NEW YORK
CHAPMAN & HALL, LIMITED, LONDON

FIFTH EDITION
Second Printing, August 1955

LIBRARY OF CONGRESS CATALOG CARD NUMBER: 54-9193

PRINTED IN THE UNITED STATES OF AMERICA

Preface

This Fifth Edition of *Steam, Air, and Gas Power* is an extensive revision which provides a textbook suitable for basic courses in heat-power theory, practice, and equipment. The intent has been to present the material in a concise and readable style. Most chapters have been rewritten and include many new diagrams, illustrations, examples, and problems; answers are given with some problems. Of special significance are the new chapters dealing with gas turbines and mechanical refrigeration.

The first two chapters are introductory and have been enlarged to explain the terminology of heat power and the arrangements of modern plants. The material on thermodynamics has been broadened. In subsequent chapters, the general-energy equation has been applied to steady-flow processes. Liquid and gaseous fuels are discussed more fully than in prior editions.

The subject of steam engines has been reduced to one chapter. The treatment of steam generators and their auxiliaries, steam turbines, steam-condensing equipment, and pumps has been modernized. Centrifugal and axial-flow compressors are included in the discussion of air and gas compression. The material on internal-combustion engines is presented with emphasis on theory and operation rather than upon description.

Grateful appreciation is expressed to all whose comments and criticisms provided a background for this revision, to those manufacturing firms who supplied illustrations and data, and to E. D. Luke, Professor of Mechanical Engineering, University of Illinois, for valuable criticism.

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Contents

CHAPTER I

Basic Definitions

1. Introduction. The use of power in modern living is so widely accepted that all students of engineering, regardless of their major interest, should have a fundamental knowledge of the subject. In any area it is necessary to define the terms used before an intelligent discussion can be made. This chapter will review such concepts as are necessary to the subject of heat power.

2. Mass, Force, and Weight. A bar of platinum iridium, designated as a standard kilogram *mass*, is kept by international agreement at the French Bureau of Standards. The standard pound mass (lb_m), 16 ounces, is defined as 0.4535924+ kilogram. Such a mass in a standard gravitational field would "weigh" 1 lb on beam or spring scales. Mass is described as that which has inertia, i.e., that which resists acceleration. A unit of mass called a *slug* is 32.174 times as large as a pound mass.

A *force* is a pull or a push, and the term implies that resistance to movement is being encountered. The unit is the pound force (lb_f) which is that force which will give an acceleration of 32.174 ft per sec per sec to a 1-lb mass. The term *pound* is a unit of either force or mass, and consequently it is indefinite unless it is specified as a pound force (lb_f) or a pound mass (lb_m). These subscripts will be used only when necessary to avoid ambiguity.

In the engineering system of units (pound mass, pound force, foot, and second) a unit conversion factor is necessary to give consistent units and dimensions. Such a *unit conversion factor* (g_c) may be determined from Newton's law and is expressed as

$$g_c = 32.174 \frac{\text{ft-lb}_m}{\text{lb}_f \text{ sec}^2} \qquad (1)$$

As used conventionally, *weight* is synonymous with mass, pound for pound. This usage is well established, and the use of weight (m) here will adhere to this meaning. Strictly speaking, weight is the force (lb_f) due to gravitational attraction for a mass. Weight is an indefinite

term since it depends on gravitational attraction which varies with location.

3. Energy. The capacity to produce an effect is called *energy*. The effect may be ever so small—a slight noise, the movement of a tiny object, wave action—yet it requires energy. Energy appears in many forms and is convertible from one form to another. Much of the subject of heat power deals with energy conversion.

Units of energy commonly used in engineering are the *foot-pound force* (ft-lb_f) and the *British thermal unit* (Btu). The foot-pound is the work done when 1 lb_f acts through a distance of 1 ft. The Btu as once defined was $\frac{1}{180}$ of the amount of energy required to raise the temperature of 1 lb of water from 32 F to 212 F. For reasons of standardization the unit was defined in 1929 by international agreement as follows:

$$1 \frac{\text{Btu}}{\text{lb deg F}} = 1 \frac{\text{int kcal}}{\text{kg deg C}} \tag{2}$$

One international kilocalorie is defined as $\frac{1}{860}$ international kilowatt-hour. From this relation it follows that 1 Btu = 778.26 ft-lb_f. This constant is called the *mechanical equivalent of heat* or Joule's constant and is designated by the symbol J.

Kinetic energy (KE) is that energy possessed by a mass due to its velocity. Any mass that is in motion is capable of producing an effect; therefore it possesses energy.

$$\text{KE} = \frac{mV^2}{2g_c} \quad \text{ft-lb}_f \tag{3}$$

where m = mass, lb_m.
V = velocity, fps.
g_c = unit conversion factor.

Potential energy (PE) is that energy possessed by a mass when it is acted on by a gravitational field. For example, a mass of 1 lb at an elevation of 100 ft in a standard gravitational field has potential energy of 1 lb_f × 100 ft, or 100 ft-lb. Since a gravitational field is the primary one encountered in heat power it is customary to think of potential energy as being equal to the product of mass in pounds and the elevation above some fixed datum.

$$\text{PE} = mZ \quad \text{ft-lb} \tag{4}$$

where m = mass which causes the force (lb_f) to exist.
Z = elevation, ft.

Internal energy (U) is that energy possessed by a mass because of its

molecular activity. The energy is stored in the form of kinetic and potential energy of the molecules. For a perfect gas the internal energy is a function of temperature only (Joule's law). For real gases, liquids, and solids the internal energy is proportional to temperature. The higher the temperature, the greater is the molecular activity and the greater is the internal energy. Absolute zero temperature is sometimes defined as a state in which molecular activity, and consequently internal energy, is zero. Internal energy is expressed in Btu per unit mass.

4. Pound Mole. A pound mole is M pounds where M is the molecular weight. It is a convenient unit of weight and volume when dealing with gases. The gram mole as used in chemistry (cgs system) is molecular weight expressed in grams. At standard conditions of temperature and pressure (0 degrees C and 760 mm) the volume of any perfect gas is 22.4 liters. The mole volume in the engineering system is larger in proportion as the pound is larger than the gram; and for conditions of 60 F and 14.7 psia, a pound mole of any gas is essentially 380 cu ft. That a mole of all gases should occupy the same volume at some specified temperature and pressure is explained by Avogadro's law, Art. 34.

5. Heat. Energy that flows by virtue of a temperature difference is called *heat*. When two bodies, one hot and one cold, are placed in proximity, with no insulating barrier between them, heat will flow to the cold body because of the temperature difference. Heat energy exists only when it is moving or flowing, since it was internal energy before leaving the first body and it is internal energy on entering the second body. The molecular activity of the first body will decrease, and that of the second will increase.

6. Temperature and Temperature Measurement. *Temperature* is an indication of the relative internal energy of a mass. For a perfect gas, temperature is a true indication of its molecular activity. A perfect gas having no internal energy would be at the lowest conceivable temperature (−459.7 F) or absolute zero. Temperature is the thermal potential that is responsible for heat flow.

The Fahrenheit scale of temperature is used in engineering work. On this scale pure water boils at 212 F and freezes at 32 F when under a standard atmospheric pressure of 14.696 psia. Its relation to the centigrade scale is shown in Fig. 1, from which it is seen that the increment between boiling and freezing on the Fahrenheit (F) scale is 180 degrees whereas on the centigrade (C) it is 100 degrees.

Therefore

$$t_f = \frac{180}{100} t_c + 32 \tag{5}$$

$$t_c = \frac{100}{180}(t_f - 32) \qquad (6)$$

where t_f = temperature, deg F.
t_c = temperature, deg C.

Absolute zero has been determined at a temperature of −459.7 F or −273.2 C (practically −460 F or −273 C). Thus absolute temperature Fahrenheit (deg F abs) is

$$\text{deg F} + 460 = \text{R} \quad \text{called degrees Rankine (deg R)} \qquad (7)$$

and absolute temperature centigrade (deg C abs) is

$$\text{deg C} + 273 = \text{K} \quad \text{called degrees Kelvin (deg K)} \qquad (8)$$

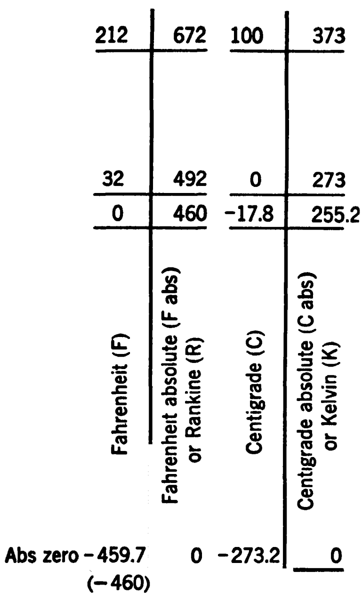


FIG. 1. Thermometer scales.

Most substances (liquid, gaseous, or solid) expand with increased temperature. If this expansion is relatively uniform over a wide range of temperatures, the substance may be used as a thermometric medium if other properties are suitable. Mercury is the most common of such substances. Since it has a very high coefficient of expansion, small changes in temperature can be read directly without need for magnification.

In 1821, Seebeck, a German physicist, observed that two wires of certain dissimilar metals, when joined together in a loop produced a current in the circuit along with an electromotive force (emf) when the junctions were at different temperatures. This is called the Seebeck effect, and the *thermocouple* operates on this principle. Lord Thompson discovered that the emf between the junctions is proportional to their temperature difference. A galvanometer placed in such a circuit will deflect in proportion to the temperature difference of the junctions.

A potentiometer, in which the emf of the thermocouple is opposed by the emf of a dry-cell battery, is used to bring the galvanometer reading to zero. Thus there is no current flowing. and the circuit resistance does not affect the reading. Temperature is then proportional to the potential developed. A standard cell is used for calibrating the dry cell. Figure 2 shows a typical potentiometer circuit.

7. Medium, Cycle, System, Boundary, and Surroundings. In all heat engines some fluid called the working fluid or *medium* such as air, steam, or ammonia passes through a series of operations—heating,

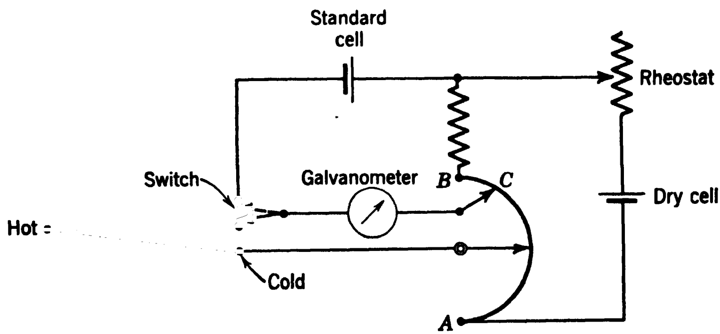


FIG. 2. Typical potentiometer circuit; (A–B) slidewire, (C) cold-junction compensator.

expansion, cooling, compression—theoretically returning each time to its original state. The same medium may be used again and again, returning at the end of each *cycle* to the same state point. Thus the same steam is used repeatedly as it travels within the closed system shown in Fig. 3.

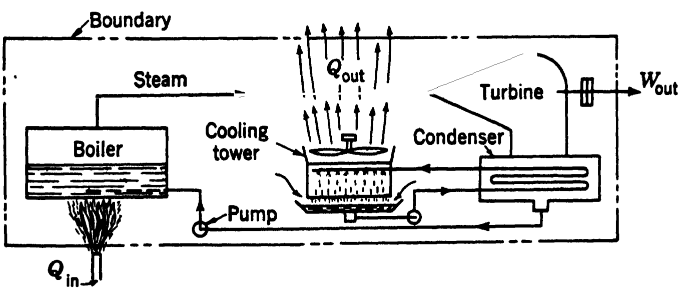


FIG. 3. A closed system as fixed by boundaries.

In Fig. 3 the *boundary* (imaginary) is so located that only work and heat flow between the system and the *surroundings*. No mass enters or leaves the system and it is called a *closed system*. In an *open system* mass as well as energy enters and leaves the system. If the boundaries in Fig. 3 were placed around the boiler only, as in Fig. 4, it would constitute an open system. *Steady flow* prevails when conditions at the boundary are nonchanging with time, i.e., when the rates of mass and

energy flow-out are equal respectively to the rates of mass and energy flow-in. Steady-flow conditions occur in many engineering operations.

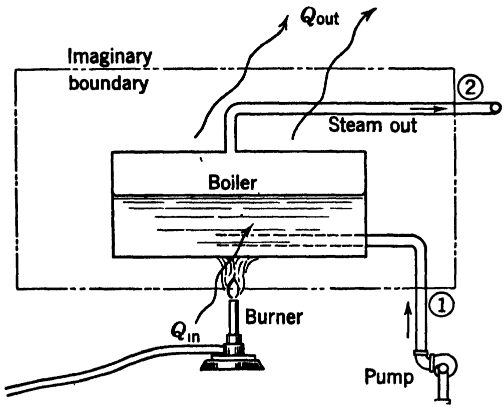


FIG. 4. An open system showing energy and mass crossing the boundary.

8. Pressure. The force per unit area exerted by a medium on its boundaries is known as *pressure* (P). In the case of gases the pressure results from the bombardment of its fixed boundaries by the molecules in their movement. Gases will normally fill the space enclosed by their real boundaries; however, if such boundaries are unrestricted as in the open atmosphere, gases are known not to fill the universe, but will have a greater density at the earth's surface because of gravitational attraction.

Pressure gages in general use are of two classes known as the Bourdon and diaphragm gages. In the Bourdon gage, Fig. 5, the pressure is exerted on the interior of a metal tube, oval in cross section, bent to fit into a circular case. The application of pressure tends to make the cross section round and thus to unroll or straighten the tube; this motion, communicated by means of links, levers, a gear sector, and a pinion, rotates a hand above a calibrated dial, and the pressure, in pounds per square inch above that of the atmosphere, may be read. In the diaphragm gage the pressure is resisted by a corrugated disk or diaphragm. The motion is transmitted to the hand in the same manner as in the Bourdon gage.

Pressures are sometimes measured in terms of a *standard atmosphere* (see Fig. 6) which is defined by international agreement as 760 mm of mercury or 29.921 in. of mercury when the mercury is at 32 F and the gravitational field is standard. The standard atmosphere is equal to 14.696 psia. *Gage pressure* is the pressure measured above atmospheric

pressure as the datum plane. The units are usually pounds per square inch gage (psig); but, in certain work, pressures are measured in inches of mercury, feet of water, or inches of water. ***Absolute pressure***, as illustrated in Fig. 6, is the pressure measured in pounds per square inch

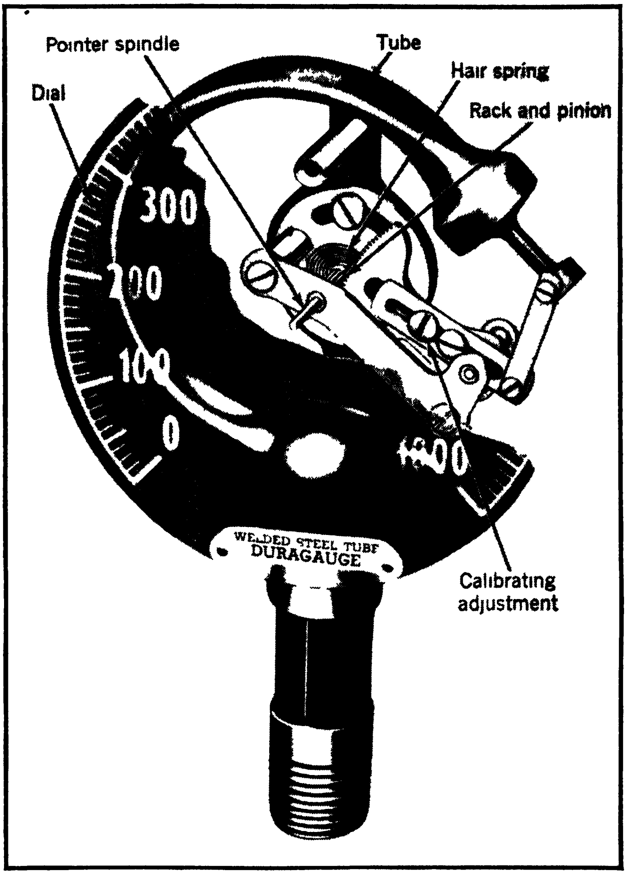


FIG. 5 Ashcroft Bourdon-type pressure gage.

above absolute zero as the datum plane (psia). It is equal to the gage pressure plus the atmospheric pressure. With atmospheric pressure as the datum plane, a ***vacuum*** is measured by the reduction in pressure below atmospheric. Thus a vacuum of 22 in. of mercury when the barometer is 29.92 means that the absolute pressure is $29.92 - 22 = 7.92$ in.

Barometers are devices for measuring the atmospheric pressure. The two principal types are the aneroid and the mercurial barometer. For

measuring low pressures U-tubes filled with mercury, water, or kerosene are commonly used. Magnification of the scale can be provided by an inclined tube. All pressure-measuring devices require occasional calibration if reliable results are expected. Maladjustment occurs from overstress, vibration, or use at too high temperatures which may cause

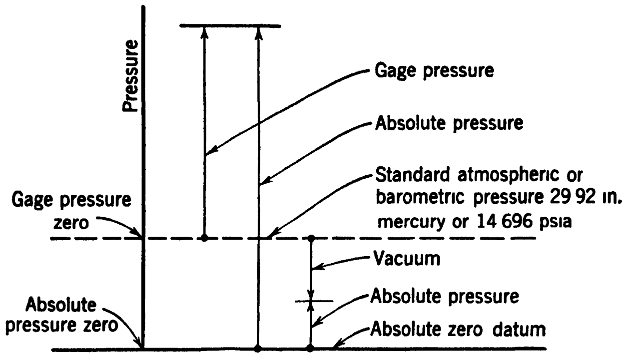


FIG. 6. Illustration of gage pressure, absolute pressure, and vacuum.

permanent set in springs or linkage. Most gages can be quickly checked for calibration by means of a dead-weight tester or against another standard or master gage.

9. Work. *Work* (W) is a form of energy and is defined as the product of a force acting through a distance in the direction of that force.

$$W = \text{Force} \times \text{Distance} \quad \text{ft-lb} \tag{9}$$

Work, like heat, is a transient phenomenon and exists only while the operation is transpiring. The energy involved has to occupy some other form immediately before and after the work has been done. There are only two significant ways by which work can be obtained from a medium (1) by the application of pressure to a movable piston (force on a movable boundary) as in all reciprocating engines and (2) by the impulse or reaction (change of momentum) of a jet of moving medium as on the blades of a turbine.

The work done per unit weight when a fluid is forced across a boundary is called *flow work*, or *flow energy*, and is the product of pressure (P) and specific volume (v). It is analogous to the work of a pump plunger in forcing water out of a cylinder. Each unit weight of fluid, as it is forced past a given point or boundary, must have flow work done on it by the plunger or by the other "layers" of fluid that are moving with it. Flow work is always based on unit weight. From

Fig. 7 the flow work

$$W = Fd$$

$$W = PAL$$

$$W = Pv \tag{10}$$

where P = pressure, psf.
A = area of pipe, sq ft.
L = flow to deliver 1 pound weight, ft.
$v = AL$ = specific volume, cu ft per lb.
W = flow work, ft-lb_f per lb_m.

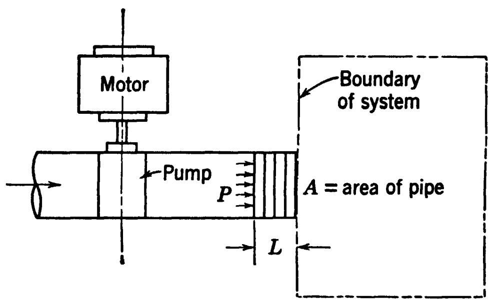


FIG. 7. Illustration of flow work.

Flow work has significance only in a flow process. In nonflow processes (closed systems) there is no flow of mass across the boundaries and consequently no flow work.

10. Enthalpy. In the flow of a fluid past a given boundary it will be noted that flow work (Pv) and internal energy (u) always appear together. It is convenient to combine these two terms for ease in computation, and their sum has been given the name "enthalpy."

$$\text{Enthalpy} = \text{Internal energy} + \text{Flow work}$$

$$h = u + \frac{Pv}{J} \quad \text{per pound weight}^1 \tag{11}$$

The definition holds for any medium and any process, flow or nonflow; however, the term has significance only for a flow process. For nonflow processes, internal energy alone is adequate, although one can evaluate $u + Pv/J$ for a fluid in which there is no flow. Thus the

[1] Note that, in general, lower-case letters are used to represent unit quantities whereas upper-case letters represent total quantities; P represents pounds per square foot and p represents pounds per square inch.

steam tables have h defined and evaluated above 32 F as a datum level, and enthalpy is tabulated for use in problems involving flow.

11. Engine Indicators. Instruments for drawing diagrams showing actual pressure-volume relations within the cylinder of an engine or a compressor are called engine indicators, Figs. 8 and 9. The indicator is attached to suitable piping leading to the clearance space in the en-

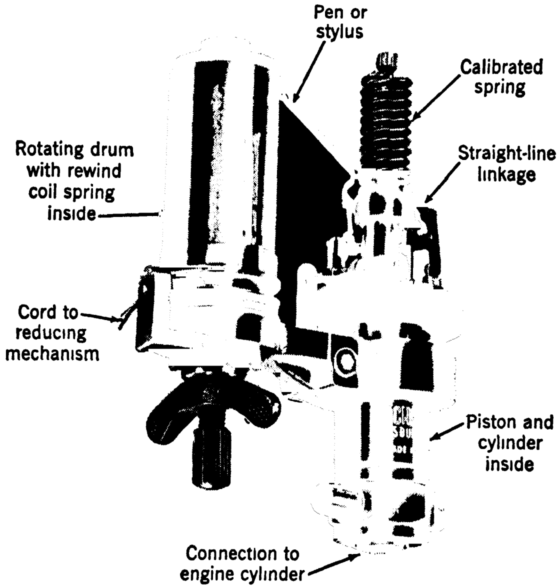


FIG. 8. Maihak standard indicator made by Bacharach.

gine cylinder. Varying pressures exerted upon the indicator piston overcome the resistance of the indicator spring in proportional amounts and cause the piston in the indicator to move. This movement is transmitted through the indicator piston rod to a linkage which gives movement to a lever carrying a pencil point or stylus which moves at all times parallel to the line of motion of the indicator piston. Indicator springs are made and calibrated to allow a vertical movement of the pencil point through a distance of 1 in. when a given pressure in pounds per square inch, known as the spring scale, is applied to the indicator piston.

The pencil or stylus makes a record upon a sheet of paper placed about the indicator drum. The drum is rotated forward and backward by a cord wound around the lower part of the drum and by the action of a coil spring within the drum. The indicator cord runs over a properly located guide pulley and is attached to some part of a re-

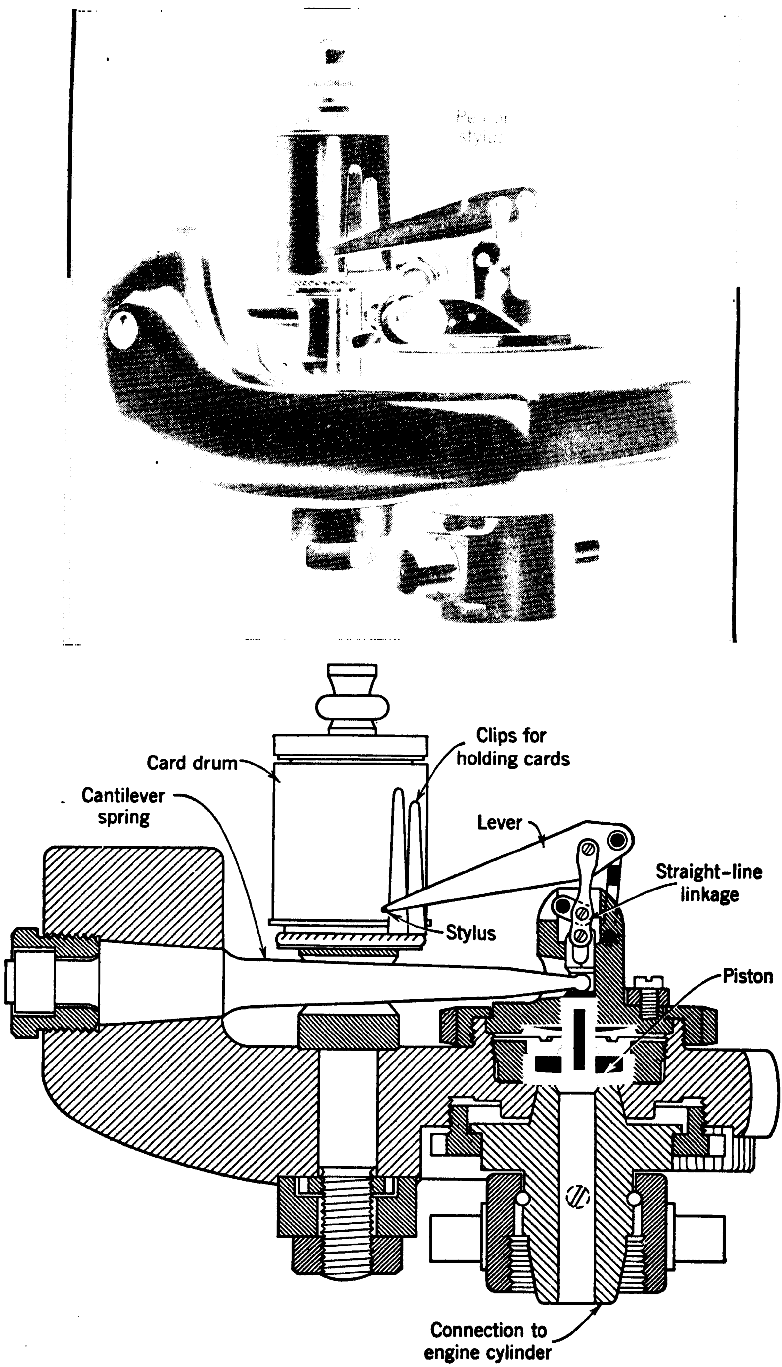


FIG. 9. Bacharach high-speed indicator.

ducing motion driven preferably by the engine crosshead. The function of the reducing motion is to reproduce exactly to a smaller scale the reciprocating motion of the engine piston. Unless the reduction of motion is exact at all points of the piston travel, the pressure-volume diagram produced by the indicator will be inaccurate.

Figure 9 shows a microindicator used for speeds as high as 2000 rpm. All the moving parts are of minimum size to reduce inertia effects. A stiff cantilever spring replaces the usual coil spring.

The work areas of indicator diagrams are usually measured by mechanical integrating devices known as polar planimeters, Fig. 10. Such

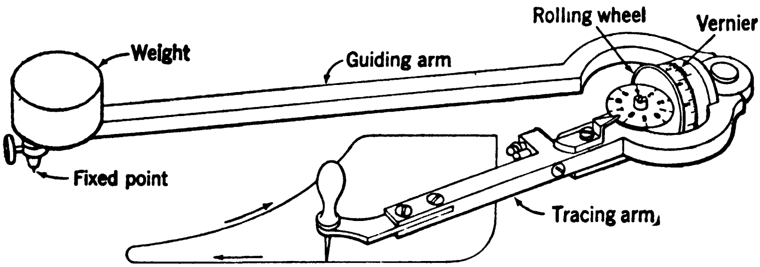


FIG. 10. Fixed-arm polar planimeter.

a mechanism, when properly constructed, adjusted, and handled, will rapidly give area measurements of sufficient accuracy. When the area of a diagram is known, the mean height or mean ordinate (O) may be determined by dividing the diagram area in square inches by the extreme length, in inches, of the enclosed area. The *mean effective pressure* (p_m) within the engine cylinder is equal to the product of the mean ordinate (O) of the indicator diagram, and the scale (S) of the spring used when the diagram was taken.

$$p_m = O \times S \quad \text{psi} \tag{12}$$

12. Power. The rate of doing work is called *power*. The engineering unit is the horsepower (hp).

$$\begin{aligned} 1 \text{ hp} &= 550 \text{ ft-lb per sec} \\ &= 550 \times 60 = 33{,}000 \text{ ft-lb per min} \\ 1 \text{ hp-hr} &= 2545 \text{ Btu} \\ 1 \text{ kw-hr} &= 3413 \text{ Btu} \end{aligned}$$

The term *indicated power* (ihp) arose from the use of an indicator for measuring the power developed within an engine cylinder. Actually

it is an expression of the power delivered by the medium to the piston.

$$\text{ihp} = \frac{p_m l A N'}{33{,}000} \tag{13}$$

where p_m = mean effective pressure, psi.
l = actual stroke of engine, ft.
A = area of piston, sq in.
N' = power strokes per minute.

Note that N' is not rpm but is a function of the number of cylinders, the engine speed, and type of cycle (two- or four-stroke cycle and single- or double-acting).

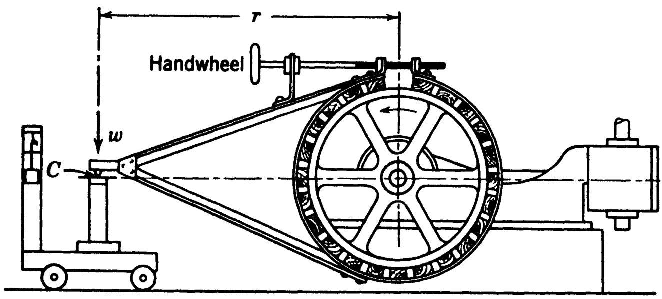


FIG. 11. Prony brake.

The actual power output delivered by the crankshaft or at the rim of the flywheel of an engine is the ***brake horsepower*** (bhp) or ***shaft horsepower*** (shp). On account of friction and windage losses occasioned by the moving parts of the engine, the brake horsepower delivered is always less than the indicated horsepower developed.

The brake horsepower of comparatively small engines is determined by means of a ***Prony brake***, Fig. 11. The brake consists of a flexible band or strap carrying a number of wooden blocks, which encircle the engine flywheel or a brake pulley mounted upon the engine shaft. The wooden blocks are in contact with the surface of the wheel, and the strap or band may be tightened or loosened upon the brake wheel by means of a hand adjustment. The horizontal distance from C to the center of the engine shaft is the length (r) of the brake arm. The Prony brake is a friction device, and the power developed at the rim of the brake wheel is dissipated as heat. Therefore, provision must be made for cooling the rim of the brake pulley.

Since work is defined as the product of a force and the distance through which the force acts, the work absorbed by a Prony brake,

Fig. 11, is

$$W = 2\pi r N w \quad \text{ft-lb per min} \tag{14}$$

where w = the net force delivered to the scale, lb.
r = length of the brake arm, ft.
N = revolutions per min.
$2\pi r N = d$ = distance that point C would travel if slippage did not occur at the band, ft per min.

Since a horsepower is defined as 33,000 ft-lb per min, the power absorbed by a Prony brake is given by

$$\text{bhp} = \frac{2\pi r N w}{33{,}000} \tag{15}$$

In equation 15 the quantity $2\pi r$ 33,000 is fixed for a given brake, and this ratio is called the *brake constant.* Most brakes are designed so that the brake constant is a simple number.

Example. Determine the length of a Prony brake arm so that bhp = $0.001Nw$.

Solution.

$$\text{bhp} = \frac{2\pi r N w}{33{,}000} = 0.001Nw$$

$$r = \frac{0.001 \times 33{,}000}{2\pi} = 5.25 \text{ ft}$$

Note that r must be expressed in feet.

The loss of mechanical power within an engine is known as *friction horsepower* (fhp). This quantity is equal to the difference of the total ihp and the bhp.. Much of this loss is due to the shearing action of the piston rings as they move against the oil film on the cylinder walls. The useful output at the shaft is further decreased by bearing friction and the power necessary to drive such auxiliaries as governor, oil pump, cooling fan, and auxiliary generator. In general, engine friction varies as the square of the speed, and the percentage of friction horsepower is greater in multicylinder engines than in simple engines.

13. Mechanical Efficiency. The term *mechanical efficiency* (e_m) is used to express the friction losses in any machine. It is the ratio between the work or power at the shaft and the corresponding work or power within the cylinder; the driving power is made the denominator.

Thus, for an engine (a driving unit)

$$e_m = \frac{\text{bhp}}{\text{ihp}} \tag{16}$$

and for a compressor (a driven unit)

$$e_m = \frac{\text{ihp}}{\text{bhp}} \tag{17}$$

Mechanical efficiency may also be expressed in terms of torque or mean effective pressure.

14. Torque. A turning effort is known as *torque* and is the product of the length of the torque arm (ft) and the net force (lb) tangent to the end of the torque arm. Torque is an important characteristic of any engine. Since it is independent of speed, it determines what must be built into a transmission between engine and driving wheels in order that the necessary turning effort may be delivered to the wheels. For example, the starting torque of an automobile engine is zero, and the engine must be disconnected from its load by means of a clutch in order that it can be cranked by a separate starting motor. The torque characteristic of a reciprocating steam engine is inherently good because it can develop maximum torque when completely stopped. Thus, a steam locomotive requires no clutch or transmission. In a Diesel locomotive the generator can be disengaged electrically from the drive wheels while the engine is started.

PROBLEMS

1. An airplane traveling 495 mph and having a mass of 4000 lb possesses how much kinetic energy?

2. A boulder weighing 1 ton rests at a point 200 ft above a datum plane. Find (*a*) its potential energy; (*b*) its kinetic energy on reaching the datum plane if it falls through a standard gravitational field without friction.

3. Express 100 F temperature as (*a*) degrees C, (*b*) degrees R, (*c*) degrees K.

4. Find the temperature at which the reading on the Fahrenheit scale is the same as the reading on the centigrade scale.

5. When the barometer is 14.696 psia, convert 10 psig to (*a*) psia, (*b*) inches of mercury absolute, and (*c*) pounds per square foot absolute.

6. When the barometer is 14.696 psia, convert an absolute pressure of 12 in. mercury to an equivalent vacuum reading in inches of mercury.

7. A jet engine exerts a thrust of 6000 lb_f in the direction of flight. Determine the work done per mile of travel.

8. One cubic foot of water is forced by a piston through an opening (boundary) against a pressure of 50 psia. Neglecting friction, determine the flow work (ft-lb).

9. A medium flowing under a pressure of 20 psia has internal energy of 200 Btu per lb and specific volume of 25 cu ft. Determine its enthalpy. 292.5 Btu, *Ans.*

10. Calculate the Btu equivalent of a horsepower-hour; of a kilowatt-hour.

11. A two-cylinder outboard motor of 2-in. bore by 2.4-in. stroke has one power stroke per cylinder for each two revolutions. If the mean effective pressure is 60 psi and the speed is 3000 rpm, find the indicated horsepower. 3.43 ihp, *Ans.*

12. Using the data of problem 11, find the mechanical efficiency and the friction horsepower if the shaft horsepower delivered to the screw (propeller) is 3.12.

13. If the length of a Prony brake arm is 2.625 ft, determine the brake constant.

14. A Prony brake is used to test a small engine. The length of the brake arm is 33 in. When the engine is running at 745 rpm, the gross load on the weighing scales is 121 lb, the tare load being 27 lb; find (*a*) brake constant; (*b*) brake torque (ft-lb); and (*c*) brake horsepower. (*a*) $5.25(10)^{-4}$, (*b*) 258.5 ft-lb, (*c*) 36.7 bph, *Ans.*

15. An engine flywheel of 4-ft diameter is running 300 rpm and driving a flat belt with a total net pull of 200 lb. Find (*a*) work done per hour; (*b*) brake torque (ft-lb); (*c*) shaft horsepower; and (*d*) indicated horsepower if the mechanical efficiency is 90 per cent.

CHAPTER 2

Heat Power Plants

15. Introduction. This chapter indicates some of the applications of heat-power equipment described in subsequent chapters. All examples discussed are for stationary land plants.

16. Types of Power Plants. Heat energy can be obtained by the proper combustion of a commercial fuel, and, when suitable equipment is provided, part of the energy thus released may be transformed into power. Each principal machine used for the conversion of heat energy into useful mechanical power output from a heat power plant is termed a *prime mover*. Examples of prime movers are steam engines, steam turbines, internal-combustion engines, and gas turbines. Each of the foregoing prime movers requires a working medium which serves as a carrier of heat energy. Steam engines and steam turbines operate with vaporized water; that is, steam under various pressures and states (Chapter 4). The gaseous products of combustion serve as the working substance in the operation of internal-combustion engines and gas turbines. Therefore, on the basis of the kind of working medium and on where the combustion of the fuel occurs, heat power plants are separated into three general classifications: *steam plants* and those powered by either *internal-combustion engines* or by *gas turbines*. Each type of plant requires a complete line of coordinated equipment for its operation.

17. Steam Power Plants. These employ either turbines or reciprocating engines, or sometimes both, not only as prime movers but as drives for auxiliary equipment, such as pumps, stokers, fans, and exciters. Steam, which is readily conveyed through pipe lines, is generated in a boiler or boilers by burning fuel in a furnace or furnaces which are an integral part of the boiler setting. The prime movers of steam power plants may be operated either *noncondensing* or *condensing*.

In noncondensing operation, Fig. 12, the steam exhausted from the prime movers is discharged at atmospheric pressure or at pressures greater than atmospheric. Condensing-plant, Fig. 13, prime movers exhaust their discharge steam into a condenser in which the pressure

is less than atmospheric and in which the steam is reduced to water. A prime mover is not able to abstract enough of the heat energy car-

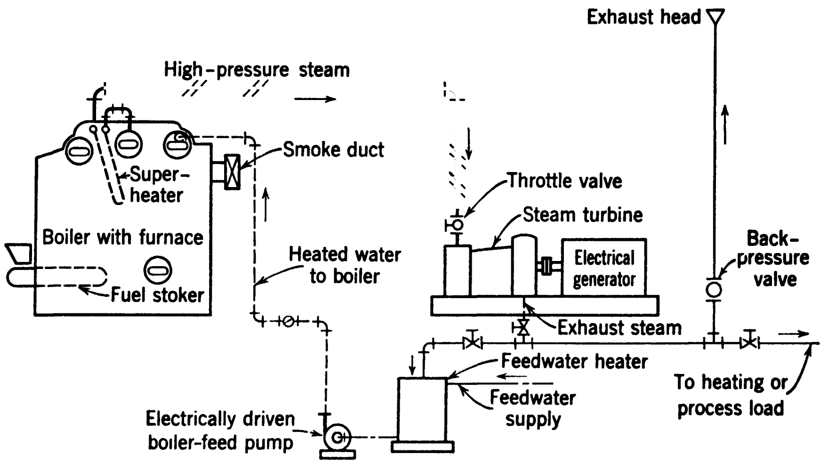


FIG. 12. Diagrammatic sketch of a simple noncondensing steam power plant.

ried by the steam to reduce it completely to water in either noncondensing or condensing operation. The principal advantages of condensing

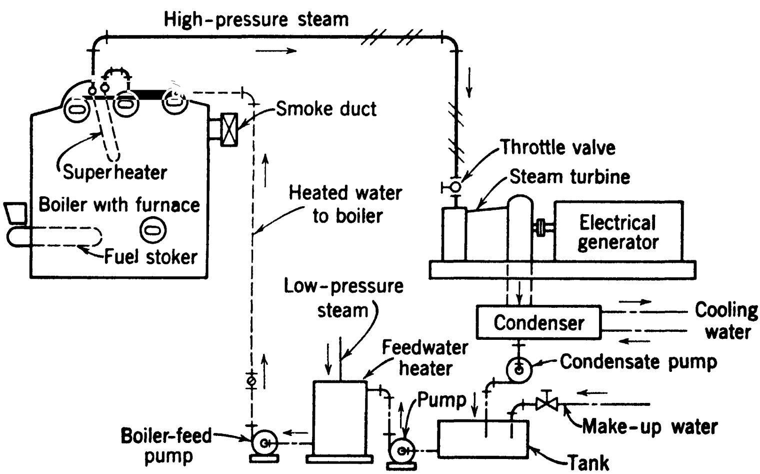


FIG. 13. Diagrammatic sketch of a simple condensing steam power plant.

operation are the increased amount of energy abstracted per pound of steam and the greater amount of power that may be developed by a given size of engine or turbine.

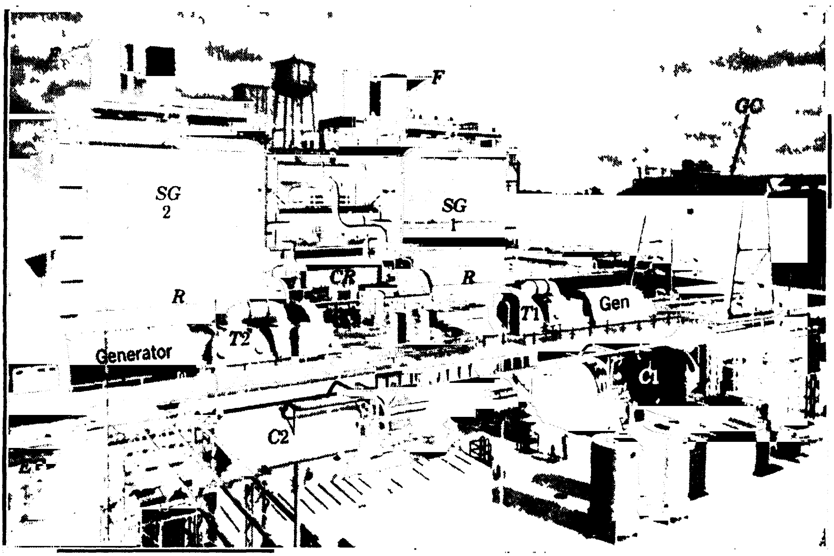
From the fuel storage to the switchboard, in each step of the process and in each machine, there are thermal and mechanical losses which reduce the amount of the available energy that is delivered as useful output. The useful output delivered by noncondensing plants may be 3 to 10 per cent of the energy contained in the fuel when the exhaust steam is discharged to the air. If the excess exhaust steam is used for heating, the percentage of the total energy of the fuel utilized in power production and heating may be increased.

The over-all efficiency, or ratio of useful energy output to the energy contained within the fuel supplied, ranges from 7 to 36 per cent for condensing plants. The highest present-day efficiencies are obtainable only in those plants that include equipment designed to secure the fullest utilization of the energy supplied. Other factors of high-efficiency operation include natural advantages (cool or cold condensing water in ample quantities, and other items) and the rigid maintenance of conditions of careful operation. The design of one ultramodern steam power plant is based on the output of a kilowatt-hour of capacity when 9303 Btu are held within the fuel supplied per unit of output.

18. Noncondensing vs. Condensing Steam Plant Operation. In each of the cases of Figs. 12 and 13 the production of superheated steam (temperature is above that normal for its pressure) occurs in the same manner. Solid fuel (coal) is fed to the boiler furnace where it is burned with the liberation of energy (heat). The automatic fuel-feeding device (stoker) is a labor-saving and helpful mechanism in the fuel-handling and combustion process. Heat energy from the burning fuel and its products of combustion is transferred to water that has been forced into the boiler by a feed pump. Steam leaves the boiler superheater at a definite pressure and is conveyed through piping to the inlet of either a steam turbine or a steam engine as the case may be. In either form of prime mover the steam is expanded with a drop in both its pressure and temperature; power thus produced is used to drive an electrical generator.

In Fig. 12 the exhaust steam is not reduced to water as it leaves the prime mover but a part of it is used for feedwater heating and part for either building heating or manufacturing-process work. Any excess or all of the exhaust steam may be discharged to the atmosphere through a back-pressure valve and an exhaust head. The back-pressure valve functions to maintain the desired low pressure in the exhaust piping. In noncondensing operation the minimum exhaust pressure may be slightly above that of the atmosphere or it may be whatever is necessary for the condition of its use. In condensing operation, Fig. 13, exhaust steam from the prime mover is reduced to its liquid (water) in

a water-cooled condenser. The reduction of steam volume produces a marked drop in the pressure of the steam leaving the power unit. A further drop of exhaust-steam pressure is secured by a device (either a vacuum pump or air ejectors) to remove air and other noncondensable gases which exert a pressure within the condenser. Air may leak into the condenser; also it may be carried by the exhaust steam along with oxygen and carbon dioxide. Under favorable conditions a steam engine



F, steam-generators stacks.	*GC*, gantry crane
SG, steam generators no. 1 and 2.	*T*, steam turbines no 1 and 2.
R, canopies over boiler-firing aisles	*C*, steam condensers no 1 and 2.
CR, control room.	*EG*, electrical switch gear.

FIG. 14. Greens Bayou outdoor steam-electric power plant near Houston, Texas.

may exhaust into a condenser vacuum of 26 in of mercury referred to a 30 in. of mercury barometric pressure. Under similar or identical initial-steam conditions a turbine is capable, under favorable conditions, of working with an exhaust pressure of 0.75 to 1.5 in. of mercury absolute. Thus the steam turbine may carry expanding steam through a greater pressure and temperature range than is possible with a steam engine and thus secure a thermal economy that is not possible with higher condenser back pressures.

19. Outdoor Steam Power Plants. The expense of a building required to house a steam power plant of great capacity is a consider-

able portion of the total cost of the plant. Reductions of the first costs of large stations have been accomplished by the construction of semi- and fully outdoor installations which are now past the experimental stage. Plants are located in this country in regions where the winters are mild and also where winter-air temperatures fall to −40 F.

Figure 14 is a view of the front and Fig. 15 is a longitudinal sectional elevation of a fully outdoor plant[1] which is capable of producing 675,000 lb per hr of steam in each of two gas-fired boilers. Each 66,000-kw turbine receives steam at 850 psig and 1000 F total temperature and is served by its individual boiler. The first two units are shown by Fig. 14; ultimately the plant will have 10 steam generators and 10 steam turbines. Portions of the boiler and turbine piping are adjacent to the steam-generator fronts shown in Fig. 14. An enclosed control room is located between the steam generators. Most of the auxiliaries (feedwater heaters, air ejectors, pumps, oil coolers, fans, and hydrogen storage tanks) are exposed to the weather, as indicated by Fig. 15. Incidentally hydrogen gas is used for cooling the electrical-generator windings.

20. Steam Plant Design Factors. The production of power by the use of steam, as a working medium, is carried on in (1) plants operating with a minimum amount of equipment; (2) those installations which serve their purposes with a few moderate-sized power units, together with the necessary auxiliaries; and (3) large stations with power-producing machines of great capacity augmented by equipment designed to secure the maximum degree of economy of operation. Chapters 6 through 13 describe and illustrate equipment which may be used in steam power plants. The design and the choice of equipment for steam power plants depend upon a number of factors. Factors to be given consideration in designing a plant and the installation of its equipment include (1) the plant location, (2) the fuel available and its cost delivered to the boiler furnaces, (3) the availability of sufficient and suitable amounts of good boiler feedwater and cooling or circulating water for condensers, (4) the magnitude and nature of the load to be handled, (5) the availability of other power service, (6) the probable necessity for future extensions to the plant, (7) the initial cost of the plant, (8) maintenance and depreciation charges, (9) the thermal economy of plant operation which is attainable, and (10) the ability of the plant to earn a fair economic return on the required investment over and above all fixed and operating charges.

[1] "The Greens Bayou Station of Houston Lighting and Power Company," by Louis Elliott, *Power Generation*, April, 1950, pp. 54–58.

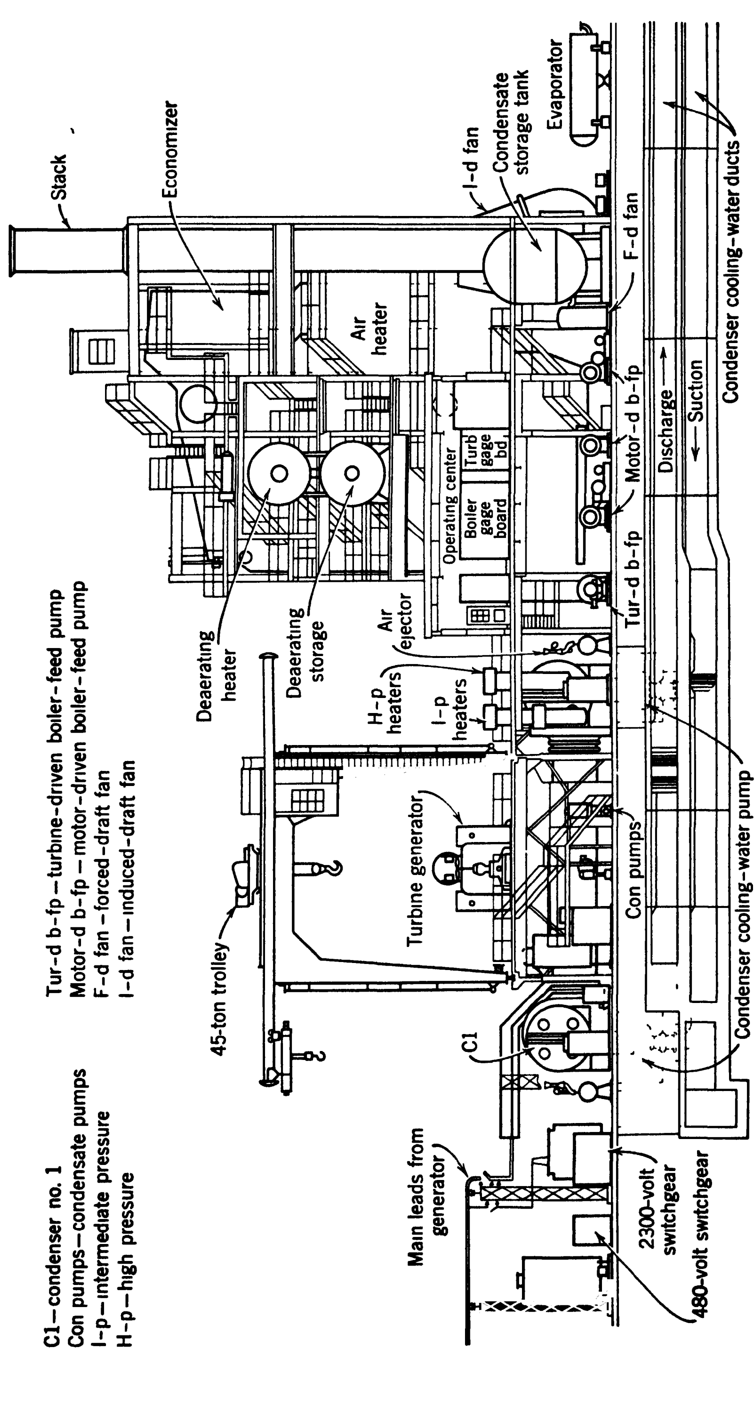


FIG 15 Longitudinal sectional elevation Greens Bayou station.

21. Methods of Improving Steam Power Plant Economy. Consideration of thermodynamic principles, which are set forth in Chapter 3, indicates that a greater efficiency of any heat-power cycle may be secured by increasing the temperature range through which the cycle is operated. Operating efficiency has been improved not only by more effective liberation and conservation of the energy of the fuels burned but also by the increase of the initial steam pressures and temperatures and by the reduction of the terminal steam pressures and temperatures to which the working medium (steam) may be expanded.

The increase of initial steam pressures and temperatures was rather gradual until the limiting exhaust pressures of 0.75 to 1.5 in. of mercury absolute were reached. The development of new steel alloys and forged steel as well as permissible welding of steel in power plant equipment has permitted the use of high-temperature high-pressure steam. At the present time most commercial steam pressures are 2500 psig (lb per sq in. gage) or less with a total temperature of 950 to 1100 F. The highest pressure contemplated at this writing is to be in a boiler and turbine unit which is to function at 4500 psig and 1150 F steam temperature in the Philo Station of the American Gas and Electric Company located near Zanesville, Ohio.

When steam pressures of 500 psig and greater, together with steam temperatures of 650 to 700 F, came into use it became necessary to employ the reheat cycle in order to avoid excessive moisture in the low-pressure stages of a compound turbine. This reheating of the steam (Art. 221) is accomplished after partial expansion of the steam in the high-pressure stages of a turbine by returning it to the reheat superheater of a boiler or through a steam-heated heat exchanger to increase the steam temperature before it enters the low-pressure stages of the turbine. As an aid in improving the plant efficiency, "bleeding" (withdrawing) of steam from various stages of turbines is employed. The bled steam may be used for regenerative feedwater heating as described in Art. 220. Another practice is the use of the "topped" or superposed plant. In this scheme of improving plant thermal efficiency, high-pressure high-temperature steam generators and turbines are superimposed upon the existing plant. When a plant is topped, the high-pressure turbine operates as a back-pressure turbine (Art. 216) to deliver its exhaust steam either to steam lines serving existing industrial equipment or to what were, prior to topping, the high-pressure steam lines of the plant. This exhaust steam together with steam derived from boilers operating at a comparable pressure then passes to the high-pressure cylinders of the original turbines where the process of heat utilization is carried out as before the plant was topped. The topping

of an existing plant necessitates boilers operating at different steam pressures and the maintenance of boiler headers with different operating steam pressures.

Applications of the topping principle have been made in new plants designed to operate on the binary vapor cycle as described in Art. 222 of Chapter 11. The commercial binary vapor installations made thus far have employed mercury vapor in the high-temperature units and steam from water in the lower-temperature phases. Mercury at 180 psia boils at 1000 F, and the resulting vapor can be expanded in a mercury turbine to produce power. The condenser of the mercury-vapor turbine serves as a boiler or heat exchanger to produce the steam used in the water portion of the cycle. The steam turbines operate with the usual steam-turbine condenser pressures and have steam bled from their stages for use in regenerative feedwater heating. The binary vapor cycle thus produced is one of the most efficient of the power cycles but its usage is subject to operating limitations.

22. Internal-Combustion Engines vs. Gas Turbines. The internal-combustion engine, in moderate-sized units, has had long usage as a prime mover. Similar uses are under way for gas turbines. In either case the combustion of the fuel takes place within the power-producing unit. The reciprocating internal-combustion engine has reached a high state of development in its various forms and has been widely employed for the production of power in stationary power plants of the smaller sizes. The gas turbine (Chapter 15), while not new in principle, is still at a stage in its development that is not equivalent to that of the reciprocating internal-combustion engine. Several manufacturers are in a position to supply gas turbines, and a number of commercial installations in oil refineries, industrial plants, locomotives, and central power stations are now in operation in the United States. Gas turbines require the continuous operation of an air compressor and also the use of metals that will withstand high working temperatures. The prospects are that these limiting requirements can be reduced in importance by improvements in air-compressor design and metallurgical processes. Most commercial gas turbines and all internal-combustion engines require fuels which are either in a gaseous or a liquid form. Investigations are solving problems in the use of pulverized solid fuels in gas turbines. This limitation does not apply to steam power plants where any fuel that is commercially available may be used as it exists in either solid, liquid, or gaseous form.

Power units of the internal-combustion type are often able to show better utilization of the heat energy of the fuel than steam plants of the same capacity, as a number of thermal losses may be avoided. When consideration is given to the use of internal-combustion equip-

ment for a power plant, attention should be directed to the following items: initial cost, maintenance charges, depreciation, and the amount of power that may be developed on a given floor space. Internal-combustion-engine and gas-turbine plants require a certain amount of auxiliary apparatus. Depending upon the kind of fuel used the extra equipment required may include fuel pumps, fuel heaters, storage tanks, gas producers, gas cleaners, coolers, scrubbers, holders, and air compressors.

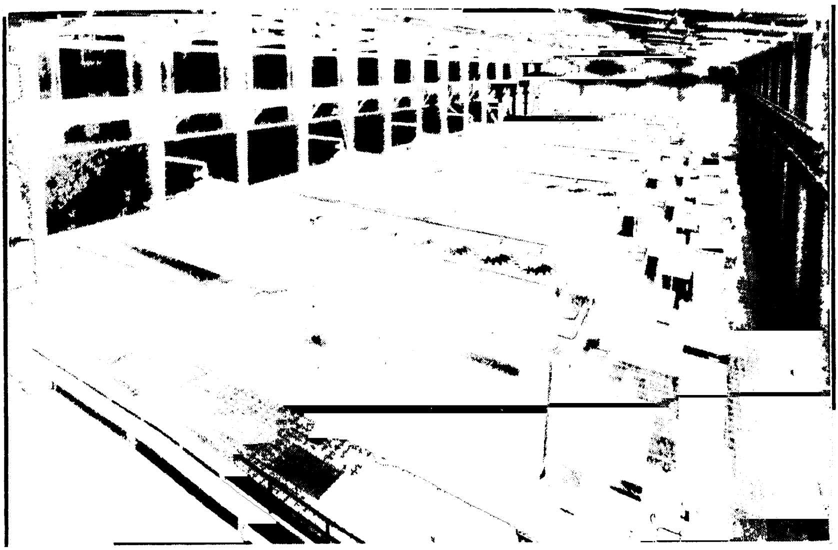


FIG. 16. Six Nordberg 2-cycle, 12-cylinder, 29 in x 40 in Diesel engines (30,000 kw at 7450 ft altitude) serving Mexico City.

23. Internal-Combustion-Engine Plants. When the combustion of the fuel takes place in one end of each of the one or more cylinders of an internal-combustion engine the unit is said to be single acting. If the process occurs in each end of each cylinder in sequence the engine is termed double acting. A further classification of such units involves the number of piston strokes required to complete a cycle in each cylinder end. Thus the engine may require two strokes (two-cycle) or four strokes (four-cycle) may be necessary. The two-stroke multicylinder engine is the one most commonly used in stationary power plants. Internal-combustion engines are not self-starting; auxiliary equipment is necessary to put them into operation. Stationary engines are usually started by means of compressed air or an electric motor.

The common fuels used in internal-combustion engines are gases and petroleum distillates ranging in density from light to heavy. All

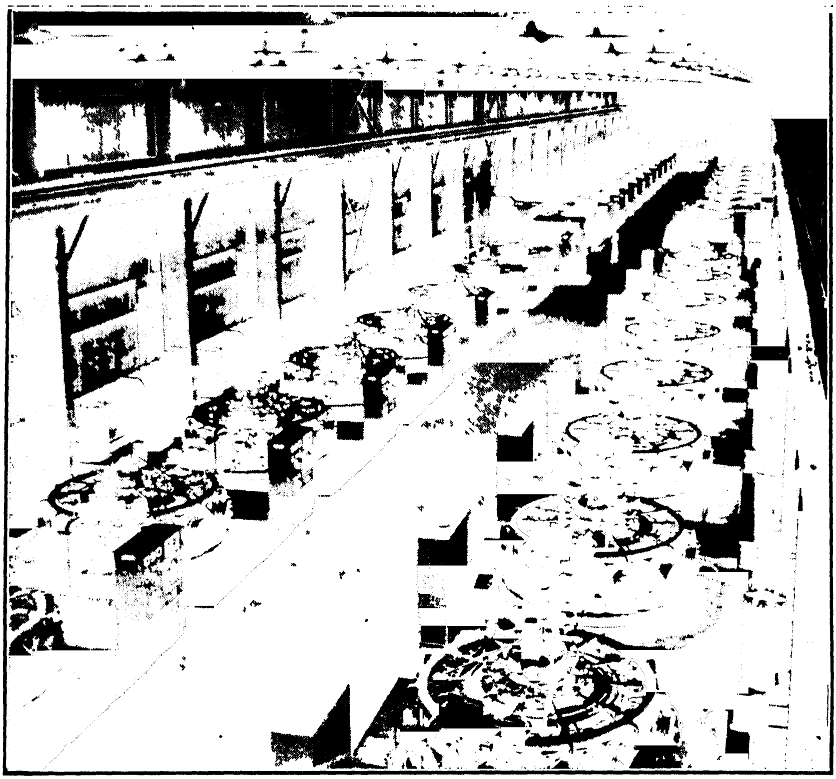


FIG. 17. Nordberg radial gas-engine generating units, 1150 kw each, at Point Comfort, Texas, Reduction Works of Aluminum Company of America. Of the 120 units, 40 are shown.

liquid fuels must be gasified before they can be burned in the air, which provides the necessary oxygen for their combustion.

Installations of two forms of two-stroke internal-combustion engines in power plants are shown in Figs. 16 and 17. Electrical generators attached to the vertical-cylinder Diesel engines are visible at the right side of Fig. 16.

The usages of engines shown in Fig. 17 are not so common as those of Fig. 16. The units of Fig. 17 have vertical crankshafts attached to electrical generators placed below the engine-room floor. Each engine of Fig. 17 has 11 cylinders placed radially about the crankshaft and develops 1650 bhp at 400 rpm. The total horsepower produced by 120 of the units, of which 40 are installed in each of 3 engine rooms, totals nearly 200,000 hp. A later installation of 12-cylinder engines of this type brings the total number at the Point Comfort, Texas, plant

to 194 units. The engines are suitable for use with natural gas, fuel oil, or dual fuels (oil and gas). A further discussion of this type of engine is given in Art. 281.

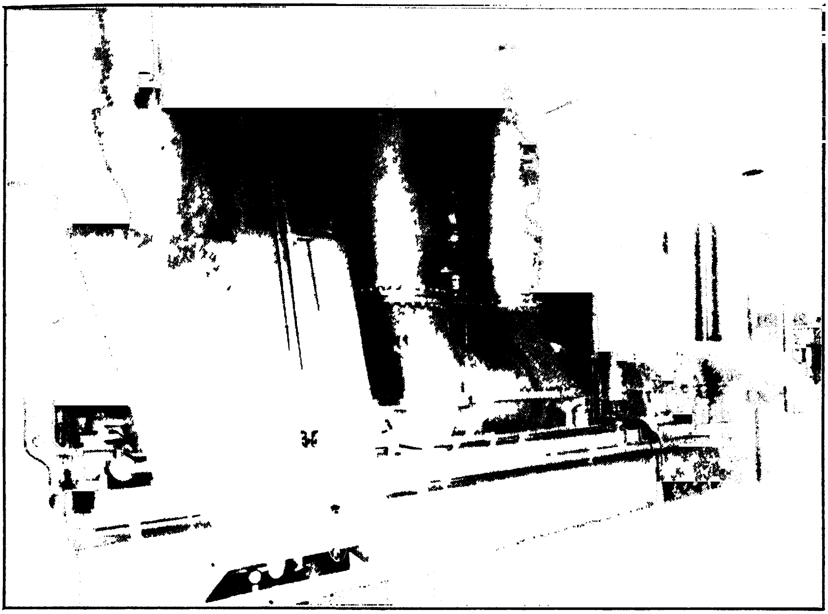


FIG. 18 Allis-Chalmers gas turbine connected to an air compressor.

24. Gas-Turbine Plants. Present-day locations of gas turbines include locomotives, industrial plants, and central-power stations. The first two commercial installations of gas turbines in this country for central-station operation are at Oklahoma City, Oklahoma, and Farmingdale, Maine. Figure 18 illustrates a gas turbine in use at an industrial plant. Different designs of gas turbines are discussed in Chapter 15. At present the temperature of the gaseous products of combustion, resulting from the burning of a fuel, is limited to about 1400 F as they enter the turbine blading. Generally the over-all thermal efficiencies of commercial turbines range from 15 to 18 per cent.

The gas turbine has these advantages over reciprocating engines of any type: (1) The heat energy obtained from a fuel is converted into power at the rotating shaft of the turbine without any reciprocating motion. (2) There are no problems of fuel injection and cylinder lubrication as with internal-combustion engines. (3) A small motor can be used to start the turbine. (4) Cooling water is not required

except where intercoolers are used between air compressors. The matter of a cooling-water supply is often of considerable importance in connection with either steam or internal-combustion-engine power plants. The gas turbine is able to operate with lower grades of fuel oils than is possible with the Diesel engine; also low-grade waste gases may be utilized as a fuel.

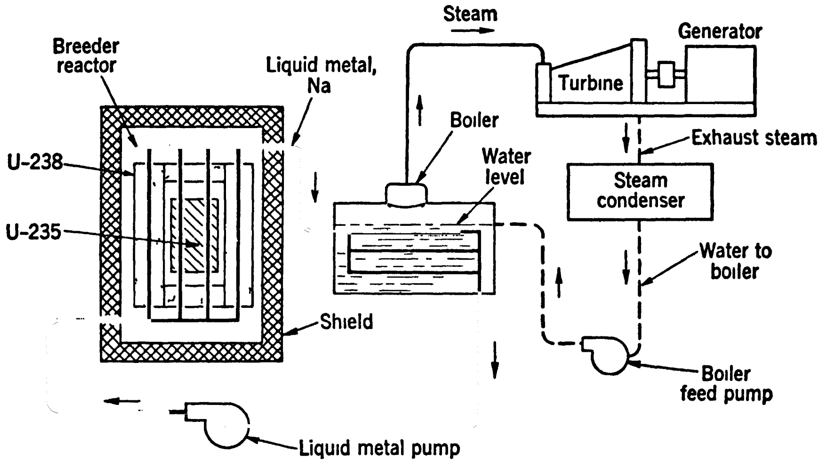


FIG. 19. Diagrammatic sketch of an atomic power plant.

25. Atomic Power Plants. The prospects are that nuclear reactors[2] can be used to release heat energy in sufficient quantities to operate steam power plant equipment of public utility system capacities.[3] The cost of power thus produced may be dependent upon the market for the fissionable products of nuclear reactors (atomic piles); it possibly may be comparable with that from a modern power plant using either coal, oil, or gas as a fuel.

Progress has been made in the use of atomic power for the propulsion of a naval submarine and a few small stationary land plants.[4] Figure 19 shows a possible layout of a small stationary power plant. The steam turbine, condenser, and water pump are similar to those commonly used in steam power plants. The unusual features of the plant

[2] "Atomic Power," by David Cochran, Sec. 17, *Kent's Mechanical Engineers' Handbook*, 12th ed., Power Vol., J. Kenneth Salisbury, editor, John Wiley & Sons, New York, 1950.

[3] "Industry Points Way to Nuclear Power. Four Reports on Technical and Economic Prospects," *Mechanical Engineering*, June, 1953, pp. 481–485.

[4] "First Atomic Power Plant," *Power Engineering*, February, 1952, pp. 54–55.

are the breeder reactor, the coolant pump[5] which serves it, and the heat-exchanger steam boiler.[6]

Figure 19 indicates the use of a liquid metal (sodium, Na) as the reactor coolant and the fluid which conveys heat to the steam boiler. Other coolants may be a mixture of sodium and potassium, NaK; helium gas, He; heavy water, D_2O; and light water, H_2O. The liquid metals sodium and potassium are satisfactory as heat-transfer agents if water does not come in contact with them to produce a violent reaction.

The energy transferred to the liquid metal, Fig. 19, is released in a breeder reactor where the fissionable uranium isotope, U-235, aids in the conversion of nonfissionable fertile uranium, U-238, into fissionable plutonium, Pu-239. The conversion takes place more rapidly than the nuclear fuel is consumed. Nuclear fuels are in either solid or liquid forms although a gaseous state is a possibility.

The design and construction of breeder reactors involve problems of great magnitude in the matter of the introduction of fuels; the removal of wastes; the use of reflectors, shields, and moderators; the prevention of components of the units from becoming radioactive in effect; the management of repair work; and the disposal of harmful atomic wastes that emit radiations harmful to human and animal life.

5 "Development of Special Pumps for Liquid Metals," by E. F. Brill, *Mechanical Engineering*, May, 1953, pp. 369–373.

6 "Liquid-Metal Heat-Transfer System for Nuclear Power Plants," by Thomas Trocki and B. D. Nelson, *Mechanical Engineering*, June, 1953, pp. 472–476.

CHAPTER 3

Principles of Thermodynamics

26. Scope. Thermodynamics is that branch of science which deals with energy transformations. All natural phenomena and physical processes involve changes of energy; hence thermodynamics plays a major role in physics, chemistry, meteorology, engineering, and other fields.

27. Property, State, and Phase. A *property* of a medium is a characteristic which is fixed when the substance is in a particular state. Conversely, the *state* of a medium is defined by fixing any two of its properties. For example, pure air at some definite temperature and pressure will have a definite volume and amount of internal energy. A property of a medium is fixed by its instantaneous state; i.e., it is independent of the process or "path" by which it arrived at that state.

Mathematically two coordinates locate a point known to lie in a given plane, as the xy plane. If a defining relation exists among three coordinates, two of which are known, the third can be computed, and the three locate a point in space. A similar relation exists with properties of matter. Any of these properties may be used as coordinates in locating a state point. The more useful of these representations are (1) the pressure-volume (PV) plane, Fig. 20(a), on which areas represent work for a reversible process (see Art. 40), and (2) the temperature-entropy (Ts) plane, Fig. 20(b), on which areas represent heat for reversible processes.

If the state of a substance can be represented by a point in the plane of a pair of coordinate axes, the properties so represented are called *point functions.* Any property of a substance that depends only upon the state and not upon the manner by which that state was attained is a point function. Variation of the magnitudes of a point function from one state to another is obtained by the difference of the initial and final conditions. Thus, a change in enthalpy (see Art. 10) can be written as $(h_2 - h_1)$. Differentials of point functions are exact; hence it is correct to write for the change in internal energy

$$dU = U_2 - U_1 \tag{18}$$

There are other magnitudes which are dependent on the path by which the change takes place and are called *path functions*. Work (W) is a performance, not a property, and is a path function. Its magnitude *cannot* be determined by the simple difference between the values

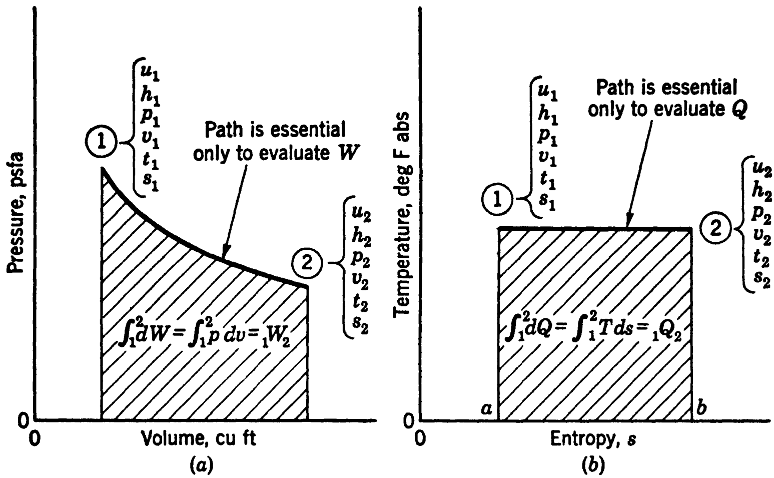


FIG. 20. Point and path functions.

of the two states. Therefore differentials of path functions are inexact. Hence

$$\int_1^2 dW = {}_1W_2 \text{[1]} \quad \text{and } not \ (W_2 - W_1) \tag{19}$$

$$\int_1^2 dQ = {}_1Q_2 \quad \text{and } not \ (Q_2 - Q_1) \tag{20}$$

The properties of a substance determine its desirability for a given purpose. Steam is well adapted to power generation and Freon to refrigeration. In traversing a cycle the medium may change from a gaseous to a liquid state and vice versa. Hence the various *phases* of a medium—solid, liquid, and gaseous—are encountered in thermodynamics.

28. First Law of Thermodynamics. The principle of the conservation of energy constitutes the first law of thermodynamics, which states that ***energy can be neither created nor destroyed, but can only be converted from one form to another.*** There are only three manifestations of en-

[1] ${}_1W_2$ is read "W from one to two."

ergy that are considered in the first law; namely, heat, internal energy, and work. If applied to the changes occurring in closed systems the first law can be written

Heat absorbed by the system =
Increase of its *internal energy* + *Work done* by the system (21)

All quantities in this equation must be measured in the same units, either Btu or foot-pounds.

A related concept states that mass cannot be destroyed but only converted from one form to another and is called the law of the conservation of mass. According to Einstein's theory of relativity, mass and energy are mutually convertible.

29. Second Law of Thermodynamics. The first law states a relation that must be satisfied in any transformation of energy. The second law gives information regarding the possibility of transformation of a given form of energy into mechanical work; this is called the availability of energy. The second law of thermodynamics has been stated in several ways. Clausius' form is: "It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature." Thus the second law may be formulated directly from the idea of temperature.

There seems to be an irreversibility in all natural processes, a universal tendency for energy to degenerate into other forms less available for transformation—an unavoidable degradation. Work may be transformed into heat by friction, and electricity is changed into heat by resistance in the wire conductor. Even when an attempt is made to convert one high-grade form of energy into another, there is an unavoidable partial transformation into low-grade forms. Griffiths in his book *Thermal Measurement of Energy* states: "Each time we alter our investment in energy, we have thus to pay a commission, and the tribute thus expected can never be wholly recovered by us and must be regarded, not as destroyed, but as thrown on the waste-heap of the Universe."

The second law states, in effect, that heat will not pass automatically from a colder to a hotter body. Heat can be forced to pass to a higher level, as in the action of a refrigeration machine, but only by applying an "external agency" to drive the machine. A heat engine acts by letting heat pass to a lower level from a hotter to a colder body not by direct conduction one to the other but by making a medium alternately take in heat from the hot body and reject heat to the cold body. Thus the medium undergoes expansions and contractions in which its pressure is, on the whole, greater during expansion than during con-

traction, and in this way *part* of the heat that is passing to a lower level through the engine is converted into mechanical work.

30. General Energy or Steady-Flow Equation. Figure 21 shows a steady-flow process for an *open system*, i.e., one in which both mass and energy may cross the boundaries. Note from Fig. 21 the various forms of energy, their symbols, and algebraic signs. Within the open system a wide interchange of energy may take place. Consider the system to

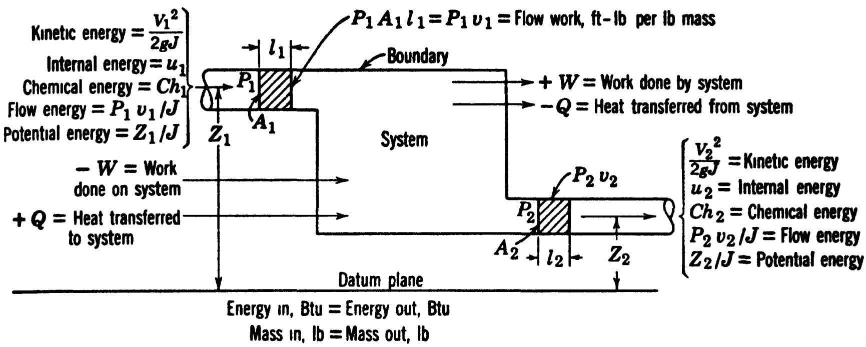


FIG. 21. Schematic representation of steady flow.

be an automotive engine with the intake manifold as the first state point and the exhaust pipe as the second state point. Within the system there would be an interchange of chemical energy in the fuel, internal energy of the gases, kinetic energy of the moving particles, flow energy, as well as heat transfer and shaft work.

From Fig. 21 it can be seen that steady flow prevails if there is no variation of flow of mass or energy with time across the boundaries of the system. Conditions within the system may be passing through cyclic or noncyclic changes. Consequently the total mass entering the system per unit of time equals the total mass leaving per unit of time; also the total energy entering the system equals the total energy leaving. The general energy equation for steady flow is

$$Q_{in} + \frac{mZ_1}{J} + \frac{mP_1v_1}{J} + mu_1 + \frac{mV_1^2}{2gJ} + \frac{W_{in}}{J} =$$

$$Q_{out} + \frac{mZ_2}{J} + \frac{mP_2v_2}{J} + mu_2 + \frac{mV_2^2}{2gJ} + \frac{W_{out}}{J} \quad (22)$$

where m = the mass flow, lb.
Q = heat transferred, Btu.
W = shaft work, ft-lb.

V = velocity, fps.
Z = elevation above datum plane, ft.
P = pressure, psfa.
u = internal energy, Btu per lb.
v = specific volume, cu ft per lb.

Chemical energy (Ch), when it is present, is included as a form of internal energy. Substituting enthalpy (h) for $(u + Pv/J)$ in equation 22 and rearranging terms,

$$Q - \frac{W}{J} = (h_2 - h_1) + \frac{V_2^2 - V_1^2}{2gJ} + \frac{Z_2 - Z_1}{J} \qquad (23)$$

where Q = net heat transferred, Btu per unit of mass flow.
W = net shaft work for the process, ft-lb per unit of mass flow.

Equation 23 is applicable for any medium in any steady-flow process, not only to rotary machines such as centrifugal fans, pumps, and compressors but also to reciprocating equipment such as steam engines and piston-type compressors.

Steady flow requires that mass must leave the open system at the rate at which it enters, or $m_1 = m_2$,
but

$$m_1 = \frac{A_1V_1}{v_1} \quad \text{and} \quad m_2 = \frac{A_2V_2}{v_2} \qquad (24)$$

where A = area of flow path, sq ft.
V = average velocity, fps.
v = specific volume of the medium, cu ft per lb.

Equation 24 is called the *continuity equation* of steady flow.

Example. In a centrifugal pump installation, Fig. 22, 100 cfm of water at 60 F are delivered by a pump having an inlet pressure of 10 psig and an outlet pressure of 200 psig, both measured at the center line of the pump. Inlet and outlet pipes are of equal diameter. Find (*a*) the work per minute done on the water and (*b*) the horsepower delivered to the water.

Solution. It is apparent that some terms are insignificant.

$$Q = 0, \text{ no heat transfer}$$

$$\frac{V_2^2 - V_1^2}{2gJ} = 0, \text{ no change in velocity}$$

$$\frac{Z_2 - Z_1}{J} = 0, \text{ no change in elevation across pump}$$

$$u_2 - u_1 = 0, \text{ no change in internal energy}$$

Therefore, using the steady-flow equation, the work per minute is

$$-W = m(P_2v_2 - P_1v_1) = mv(P_2 - P_1)$$

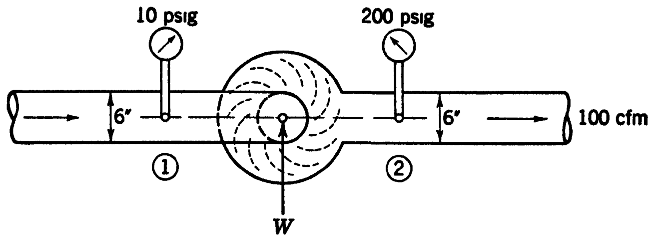


FIG 22 Steady flow in centrifugal pump

which is the difference in energy (see Art 24) across the pump

$$mv_1 = mv_2 = V_1 = V_2 = 100 \text{ cu ft}$$

(a) $$-W = 100[(144 \times 200) - (10 \times 144)]$$

$$-W = 2{,}736{,}000 \text{ ft-lb per min, done on the water}$$

(b) The horsepower delivered to the water is

$$\frac{2{,}736{,}000}{33{,}000} = 82\ 9$$

31. Simple Energy Equation for Nonflow Process. In a closed system, Fig 23, mass flow does not cross the boundaries, consequently

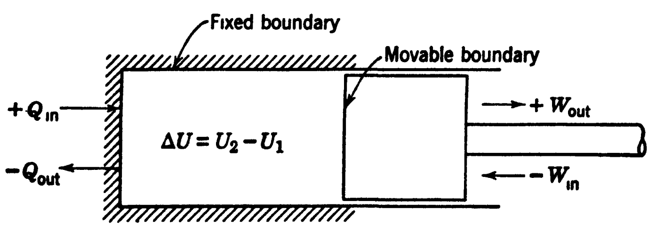


FIG 23 Nonflow or closed system.

the energy terms associated with mass movement (kinetic, potential, and flow energy) are zero, and the general energy equation simplifies to

$$Q = (U_2 - U_1) + \frac{W}{J} \tag{25}$$

where Q = heat supplied, Btu.

$U_2 - U_1$ = change in internal energy, Btu.

$\frac{W}{J}$ = work done by the system, Btu.

Equation 25 is called the *nonflow equation* or the *simple energy equation* and is an expression of the first law of thermodynamics or the law of conservation of energy for such systems.

Figure 23 shows that with such a limited system a balance must be made for heat, work, and change in internal energy. If heat is added and no work is done the internal energy must increase. If no heat is added and work is done, the internal energy must decrease. Finally, if heat is added and work is done, the nonflow equation becomes, for a small change in the system,

$$dQ = dU + \frac{dW}{J} \tag{26}$$

Since dQ and dW are inexact differentials and dU is exact, the integration of equation 26 is

$$_1Q_2 = U_2 - U_1 + \frac{_1W_2}{J} \tag{27}$$

32. Nonflow Work. In Fig. 24 the medium contained within the cylinder constitutes a closed system. The medium is capable of doing work or having work done on it by the movement of the piston (a boundary). Thus a force can be made to act through a distance in the direction of that force, and work will be done. *The work is considered positive if done by the medium and negative if done on the medium.* Assuming the frictionless piston in Fig. 24 to be moving from point c to point d, the gas pressure starting at point 1 will follow some path as indicated to point 2. At some point x let the pressure on the face of the piston be P, while the piston is moved a distance dL, this distance being infinitely small so that P may be considered constant during the movement. If the area of the piston is A, the total force on it is PA and the work done during this increment is $PA\ dL$, but $A\ dL$ is dV, a small change in volume; hence

$$dW = P\,dV$$

If this equation is integrated between limits such as 1 and 2,

$$_1W_2 = \int_1^2 P\,dV \tag{28}$$

Equation 28 is the general expression for nonflow work with the restriction that all friction effects are neglected. In Fig. 24, $_1W_2$ will be a negative number indicating work done *on* the medium. This work is graphically given by the area 1–2–*d*–*c*–1 on the PV plane and is the work of compression. It is important to note that areas on the

PV plane represent work for reversible processes, also that pressures are based on the absolute values (not atmospheric) and in pounds per square foot (not per square inch).

If heat energy is added at point 2 the pressure will rise to some value such as that at point 3. No work will be done between points 2 and 3

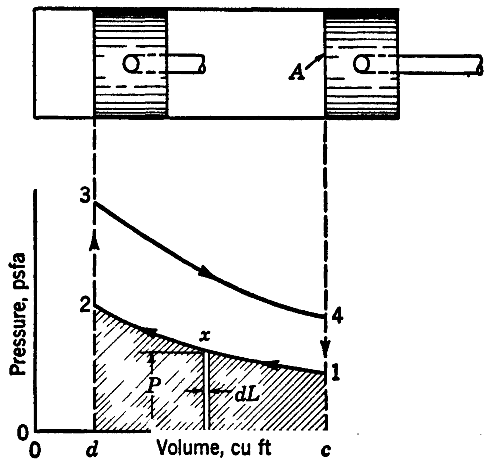


FIG. 24 PV diagram illustrating nonflow work.

since dV is zero. If then the piston is allowed to return from d to c by some path 3 to 4 as indicated, the work done will be ${}_3W_4 = \int_3^4 P\,dV$, and is shown graphically by the area 3–4–c–d–3. The value of ${}_3W_4$ will be a positive number indicating work done by the medium.

If the medium is allowed to cool from point 4 to point 1 while the piston is at point c, a cycle (Otto) will have been completed. The net work obtained will be the algebraic summation of the work generated around the cycle, or

$$\begin{aligned} W_{net} &= {}_1W_2 + {}_2W_3 + {}_3W_4 + {}_4W_1 \\ &= \int_1^2 P\,dV + 0 + \int_3^4 P\,dV + 0 \qquad (29) \\ &= \text{area } (1\text{–}2\text{–}3\text{–}4\text{–}1) \end{aligned}$$

33. Perfect Gas Laws. A *perfect gas* is one that obeys exactly the laws of Boyle and Charles, also Joule's law (Art. 36) and Avogadro's law (Art. 34). In reality there is no perfect gas; however, air, oxygen, nitrogen, helium, and several others behave sufficiently like perfect

gases to be treated as such. Any gas approaches this ideal state as its temperature is increased and its pressure is reduced; i.e., as a gas becomes superheated or is removed from that state where it may be condensed into liquid. Gases close to the liquid phase are called *vapors*.

BOYLE'S LAW. At constant temperature, the volume of a given weight of a perfect gas varies inversely as the absolute pressure; i.e.,

$$P_1 V_1 = P_2 V_2 = P_3 V_3 = C \tag{30}$$

where P = absolute pressure, psfa.
V = volume of gas, cu ft.
C = a constant.

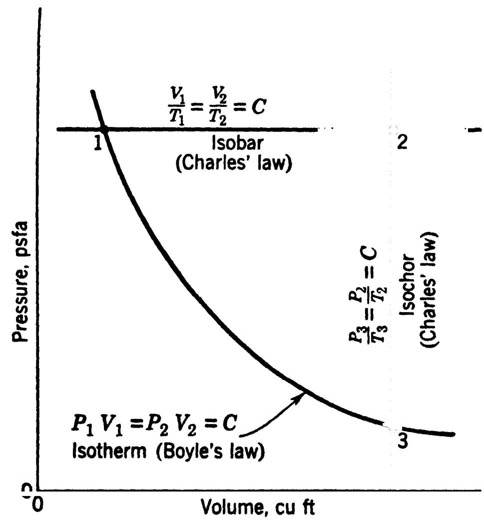


FIG. 25. Perfect gas relations.

A curve connecting state points having the same temperature is called an *isotherm* of the gas and is an equilateral hyperbola when plotted on the PV plane, Fig. 25.

CHARLES' LAW. When a given weight of a perfect gas receives or rejects energy while being held at constant volume, the absolute pressure varies directly as the absolute temperature, Fig. 26(a); thus $P/T = C$

or
$$\frac{P_1}{T_1} = \frac{P_2}{T_2} = \frac{P_3}{T_3} = \cdots = \frac{P_n}{T_n} \tag{31}$$

A curve connecting state points having the same specific volume is called an *isochor*. Isochors of a perfect gas are vertical straight lines on a PV plane, Fig. 25.

Another part of Charles' law states that, when a given weight of a perfect gas receives or rejects energy while being held at a constant pressure, the volume varies directly as the absolute temperature, Fig. 26(*b*); thus $V/T = C$

or $$\frac{V_1}{T_1} = \frac{V_2}{T_2} = \frac{V_3}{T_3} = \cdots = \frac{V_n}{T_n} \tag{32}$$

A curve connecting state points having the same pressure is called an *isobar*. Isobars of a perfect gas are horizontal straight lines on a PV plane, Fig. 25.

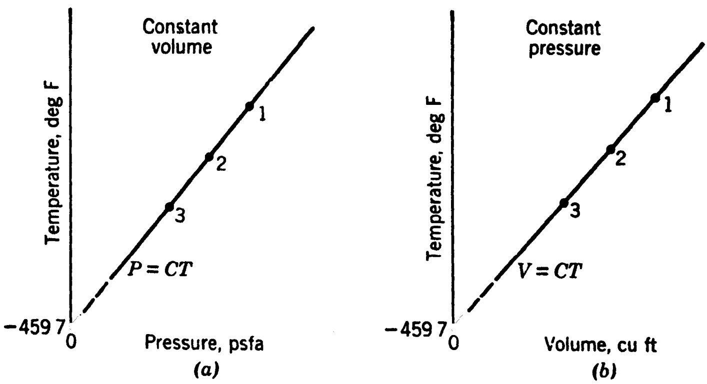


FIG. 26 Graphical representation of Charles' law.

34. Characteristic Equation of a Perfect Gas. The laws of Boyle and Charles are graphically shown in Fig. 25. Let points 1 and 2 be selected arbitrarily; through point 1 draw an equilateral hyperbola and through point 2 a vertical straight line. Mark their intersection at point 3. Boyle's law applies to points 1 and 3; hence, assuming any fixed weight of gas,

$$P_1V_1 = P_3V_3$$

since $T_1 = T_3$. Dividing these equations gives

$$\frac{P_1V_1}{T_1} = \frac{P_3V_3}{T_3} \tag{33}$$

Charles' law applies to points 2 and 3. Then

$$\frac{P_2}{T_2} = \frac{P_3}{T_3} \tag{34}$$

since $V_2 = V_3$. Therefore,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3} \tag{35}$$

irrespective of how points 1 and 2 were chosen. Equation 35 shows that for a perfect gas PV/T has the same value at every state point for a fixed weight.

For a weight of 1 lb,

$$\frac{Pv}{T} = R \tag{36}$$

where R = the individual gas constant, ft-lb_f per (lb_m)(deg F abs).
v = specific volume, cu ft per lb.
P = pressure, psfa.
T = temperature, deg F abs.

Equation 36 would be true in any system of units, and those used above are engineering units. The individual gas constant R has a definite value in these units for each gas (see Table I). If each member of equation 36 is multiplied by mT,

$$Pvm = mRT$$

or

$$PV = mRT \tag{37}$$[2]

where $V = vm$ = volume of m pounds, cu ft.
Equation 37 is called the *characteristic equation* of a perfect gas. For problems involving two states the homogeneous form of the equation is best adapted for calculation purposes; i.e.,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} \tag{38}$$

with P, V, and T as previously defined.

Example. Find the value of R for air.

Solution. Let m = unit weight, lb. Then

$$v = 12.389 \text{ cu ft when } T = (32 + 459.7) \text{ deg F abs}$$

$$P = (14.696 \times 144) \text{ psfa}$$

$$R = \frac{PV}{mT} = \frac{14.696 \times 144 \times 12.389}{1(32 + 459.7)} = 53.35$$

[2] An equation of state which was proposed by van der Waals for representing the P-V-T relations of actual (imperfect) gases and vapors is

$$P = \frac{RT}{V - b} - \frac{a^2}{V^2}$$

where a, b, and R are constants.

TABLE I

THERMAL PROPERTIES OF GASES*

Gas		Molecular Wt, lb_m per mole	R, ft-lb_f per (lb_m) (deg F abs)	c_p, Btu per (lb_m) (deg F)	c_v, Btu per (lb_m) (deg F)	k, c_p/c_v	B, ft-lb_f per (mole) (deg F abs)
Air		28.97	53.35	0.240	0.171	1.40	1545
Ammonia	(NH_3)	17.02	90.70	0.520	0.404	1.29	1542
Argon	(A)	39.90	38.73	0.125	0.075	1.67	1544
Carbon monoxide	(CO)	28.00	55.18	0.249	0.178	1.40	1543
Carbon dioxide	(CO_2)	44.00	35.12	0.202	0.156	1.29	1536
Gasoline	(C_8H_{18})	114.14	13.54	0.407	0.390	1.04	1546
Helium	(He)	4.00	386.00	1.250	0.753	1.66	1544
Hydrogen	(H_2)	2.02	766.60	3.445	2.460	1.40	1546
Methane	(CH_4)	16.03	96.40	0.532	0.408	1.30	1541
Nitrogen	(N_2)	28.02	55.16	0.248	0.177	1.40	1543
Oxygen	(O_2)	32.00	48.30	0.219	0.157	1.39	1544
Steam	(H_2O)	18.02	85.60	0.445	0.335	1.33	1542

* Average values for moderate temperatures.

Example. Find the volume of 30 lb of air at 70 F under a pressure of 90 psig, when the barometric pressure is 29.92 in. of mercury.

Solution. $PV = mRT$

Therefore
$$V = \frac{mRT}{P} = \frac{30 \times 53.35 \times (70 + 459.7)}{144(90 + 14.696)} = 56.22 \text{ cu ft}$$

Example. If the air in the preceding example is expanded to a volume of 284 cu ft and a pressure of 5 psig, what would be the final temperature?

Solution.
$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Therefore
$$\frac{144 \times (90 + 14.696) \times 56.22}{459.7 + 70} = \frac{144 \times (5 + 14.696) \times 284}{T_2}$$

$$T_2 = 503.4 \text{ deg F abs}$$

or
$$t_2 = 503.4 - 459.7 = 43.7 \text{ F}$$

If each member of equation 36 is multiplied by MT where M is the molecular weight of a given gas,

$$PvM = MRT$$

or
$$PV' = MRT \tag{39}$$

where V' is the specific mole volume and contains M pounds weight, M being the molecular weight

From Avogadro's law—*equal volumes of all gases at the same temperature and pressure contain the same number of molecules*—it follows that

$$\frac{m_1}{m_2} = \frac{M_1}{M_2} \tag{40}$$

i.e., the weight (m_1) of any gas is to that of another (m_2) as their molecular weight M_1 and M_2 when each occupies the same volume at the same pressure and temperature.

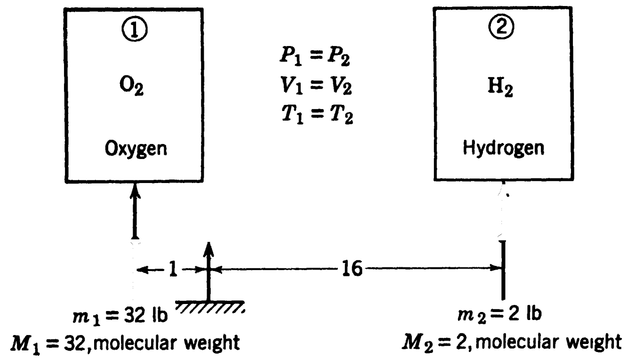


FIG. 27. Mole volumes. (Both containers have the same number of molecules, i.e., 1 mole; Avogadro's law)

Using equation 37,

$$PV = mRT \quad \text{or} \quad m = \frac{PV}{RT} \tag{41}$$

and Avogadro's law,

$$\frac{M_1}{M_2} = \frac{m_1}{m_2} = \frac{P_1V_1/R_1T_1}{P_2V_2/R_2T_2} = \frac{R_2}{R_1} \tag{42}$$

Therefore,

$$M_1R_1 = M_2R_2 = B \tag{43}$$

or the product of the molecular weight (M) of a gas and its individual gas constant R is a new constant B which is the same for all perfect gases and essentially so for real gases. Any two gases could have been used to illustrate this relation as shown in Fig. 27.

The equation $PV' = MRT$ may now be written $PV' = BT$, which is similar in form to equation 36, but on a mole basis. Multiplying both members by N moles,

$$PV'N = NBT$$

which may be simplified to

$$PV = NBT \tag{44}$$

where P = pressure, psfa.
V = volume of N moles, cu ft.
N = number of moles.
T = temperature, deg F abs.
B = universal gas constant, ft-lb per (mole)(deg F abs).
V' = specific mole volume, cu ft.

Equation 44 is the characteristic equation on a mole basis and is easier to use than equation 37 since the universal gas constant B, Table I, is essentially the same for all perfect gases; i.e.,

$$B = \frac{PV}{NT} = 1545 \text{ ft-lb per (mole) (deg F abs)} \tag{45}$$

or

$$\frac{1545}{778.26} = 1.986 \text{ Btu per (mole) (deg F abs)}$$

Example. Determine the volume of 1 mole of a gas at 60 F and 14.696 psia.

Solution.

$$V = \frac{NBT}{P} = \frac{1 \times 1545 \times (60 + 459.7)}{14.696 \times 144} = 379 \text{ cu ft}$$

In the engineering system the volume of 379 cu ft corresponds to 22.4 liters in the cgs system and is called a standard pound mole volume.

Example. A tank contains 10,000 cu ft of methane gas (CH_4) at a temperature of 60 F and a pressure of 20 psia. Find (*a*) the number of moles and (*b*) the weight of gas present.

Solution. $PV = NBT$

$$V = \frac{NBT}{P} = \frac{1 \times 1545 \times 519.7}{144 \times 20} = 278 \text{ cu ft per mole}$$

(*a*) $$\frac{10{,}000}{278} = 35.97 \text{ moles of } CH_4 \text{ in tank}$$

(*b*) $$35.97 \times 16 = 575.5 \text{ lb of } CH_4 \text{ in tank}$$

Since $B = MR = 1545$ ft-lb per (mole) (deg F abs), R (the individual gas constant for that gas) is B/M. From Table I, M for air is 28.967; therefore, R for air is $1545/28.967 = 53.35$ ft-lb_f per (lb_m) (deg F abs).

From Avogadro's law, Fig. 27, it may be seen that, for gases at the

same temperature and pressure,

$$\frac{M_1}{M_2} = \frac{\rho_1}{\rho_2} = \frac{v_2}{v_1} = \frac{1/v_1}{1/v_2} \tag{46}$$

where ρ = density, lb per cu ft.
v = specific volume, cu ft per lb.
M = molecular weight.

35. Dalton's Law. In many processes a mixture of gases is involved. Dalton's law states that *in a mixture of perfect gases each gas exerts a partial pressure P that is independent of the other components, and the total pressure P_T of a mixture of gases is equal to the sum of the partial pressures*; thus

$$P_T = P_1 + P_2 + P_3 + \cdots + P_n \tag{47}$$

Other relations for mixtures of perfect gases can be stated. The total weight of a mixture of perfect gases is equal to the sum of the weights of the components; thus

$$m_T = m_1 + m_2 + m_3 + \cdots + m_n \tag{48}$$

Also, each component at its partial pressure occupies the total volume; thus

$$V_T = V_1 = V_2 = V_3 = \cdots = V_n \tag{49}$$

Likewise, the total number of moles N_T is the sum of the moles of the components; thus

$$N_T = N_1 + N_2 + N_3 + \cdots + N_n \tag{50}$$

The *mole fraction* (x) of any one gas in a mixture is, therefore,

$$x_1 = \frac{N_1}{N_1 + N_2 + N_3 + \cdots + N_n} = \frac{N_1}{N_T} \tag{51}$$

By use of $P_1V_1 = N_1BT_1$, which applies to any one gas in the mixture, and $P_TV_T = N_TBT_T$, which applies to the total mixture, the following relation is obtained:

$$\frac{P_1V_1}{P_TV_T} = \frac{N_1BT_1}{N_TBT_T} \quad \text{or} \quad \frac{P_1}{P_T} = \frac{N_1}{N_T} \tag{52}$$

or

$$P_1 = \frac{N_1}{N_T} P_T = x_1P_T \tag{53}$$

Therefore, the partial pressure of any one gas is equal to its mole fraction times the total pressure of the mixture.

Another relation for a mixture of perfect gases, known as Amagat's law, states that, *if each component in a mixture of gases could be segregated at the same temperature and pressure as the mixture, the sum of the separate volumes would equal that of the original mixture, and the partial pressure of each component would equal the total pressure* (see Fig. 28).

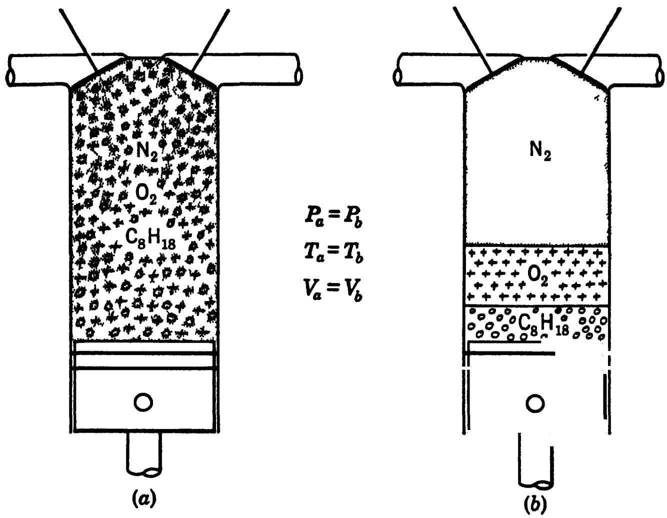


FIG 28 Illustration of Amagat's law for a mixture of air and gasoline.

Applying $PV = NBT$ to the components of a mixture,

$$\frac{P_1V_1}{P_TV_T} = \frac{N_1BT_1}{N_TBT_T} \quad \text{or} \quad \frac{V_1}{V_T} = \frac{N_1}{N_T} = x_1 \tag{54}$$

or

$$V_1 = \frac{N_1}{N_T} V_T = x_1 V_T \tag{55}$$

Therefore the volume of any one component in a mixture of gases is equal to the mole fraction times the total volume.

36. Joule's Law: Internal Energy of Perfect Gases. In 1844, Joule, an English physicist, compressed air to a pressure of about 300 psig in a vessel A, as in Fig. 29(a), which was connected to another vessel B through a valve C. Vessel B was evacuated, and the entire system immersed in a bath of water at room temperature. When thermal equilibrium was established the valve was opened and the thermometer observed. No change in temperature was noted by Joule.

The process was adiabatic, $Q = 0$ (no change in water temperature); no work crossed the boundaries, $W = 0$; and it was a nonflow system; therefore,

$$Q - W = U_2 - U_1 = 0$$

or

$$U_1 = U_2$$

From this, Joule concluded that *the internal energy of a perfect gas is a function of temperature only*, since the pressure had decreased and the volume had increased while the temperature and internal energy remained constant.

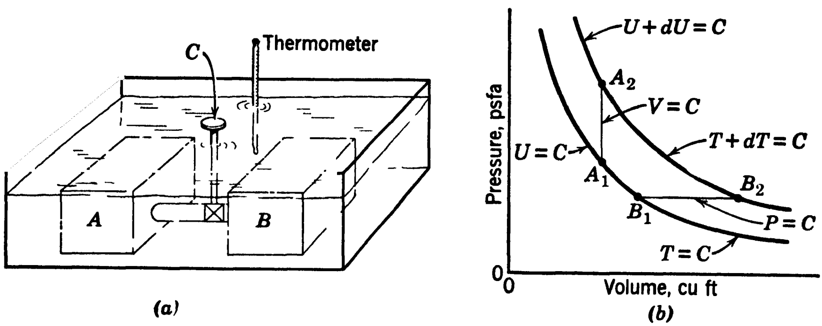


FIG. 29. Illustration of Joule's law.

More accurate measurements show that for real gases a slight drop in temperature does occur; however, Joule's conclusion was correct for perfect gases, and it is sufficiently correct for real gases at low pressures to yield satisfactory solutions to many problems. Joule's law is a criterion for defining a perfect gas. Mathematically, if a gas obeys Boyle's and Charles' laws it must also obey Joule's law.

37. Specific Heat. If heat is absorbed by a substance there is a change in temperature. The ratio of the heat absorbed to the change in temperature is called the heat capacity. If a unit mass of substance is considered, the measure of the amount of heat absorbed in order to change the temperature 1 deg is called the *specific heat* of the substance. A better name than specific heat might be specific thermal capacity, since heat has been defined as energy in transition by virtue of a temperature difference.

The *mean specific heat* (c_m) of a substance over a given range is the average quantity of heat required to raise the temperature of a unit mass of the substance 1 deg under the existing conditions of volume and

pressure. In practically all engineering work, the mean specific heats are used.

Table I lists mean values of certain thermal data for several gases; constancy of the values given may be assumed without serious error for moderate temperatures (up to 300 F).

By definition, for unit mass,

$$c = \frac{dq}{dT} \quad \text{or} \quad dq = c\,dT \tag{56}$$

and, for m pounds,

$$_1Q_2 = m\int_1^2 c\,dT \tag{57}$$

where c = specific heat,[3] Btu per (lb) (deg F).

$_1Q_2$ = energy absorbed from state 1 to state 2, Btu per lb.
m = mass, lb.
dT = temperature change, deg F.

Since the addition of energy to a given mass can produce effects other than an increase in temperature, it is necessary to define the conditions under which the energy is added. Two special cases are important; first, when the mass is held at constant volume

$$_1Q_2 = m\int_1^2 c_v\,dT \tag{58}$$

and second, when the mass is held at constant pressure

$$_1Q_2 = m\int_1^2 c_p\,dT \tag{59}$$

where c_p = specific heat at constant pressure, Btu per (lb) (deg F).
c_v = specific heat at constant volume, Btu per (lb) (deg F).

Assume that the curves shown on the PV plane in Fig. 29(b) are two neighboring isotherms for a perfect gas, one at constant temperature T and the other at constant temperature $(T + dT)$. According to Joule's law the internal energy has the same value U at every point on the first isotherm, regardless of pressure or specific volume, and a value $(U + dU)$ at every point of the second curve. Hence any path con-

[3] The term "specific heat" originated while the Btu was defined as the amount of heat required to raise the temperature of 1 lb of water 1 deg F at the average temperature of 60 F. Specific heat was then a comparison of the thermal capacity of a given mass to that of an equal mass of water. Since the Btu is no longer defined in terms of water, the thermal capacity can be expressed directly in Btu.

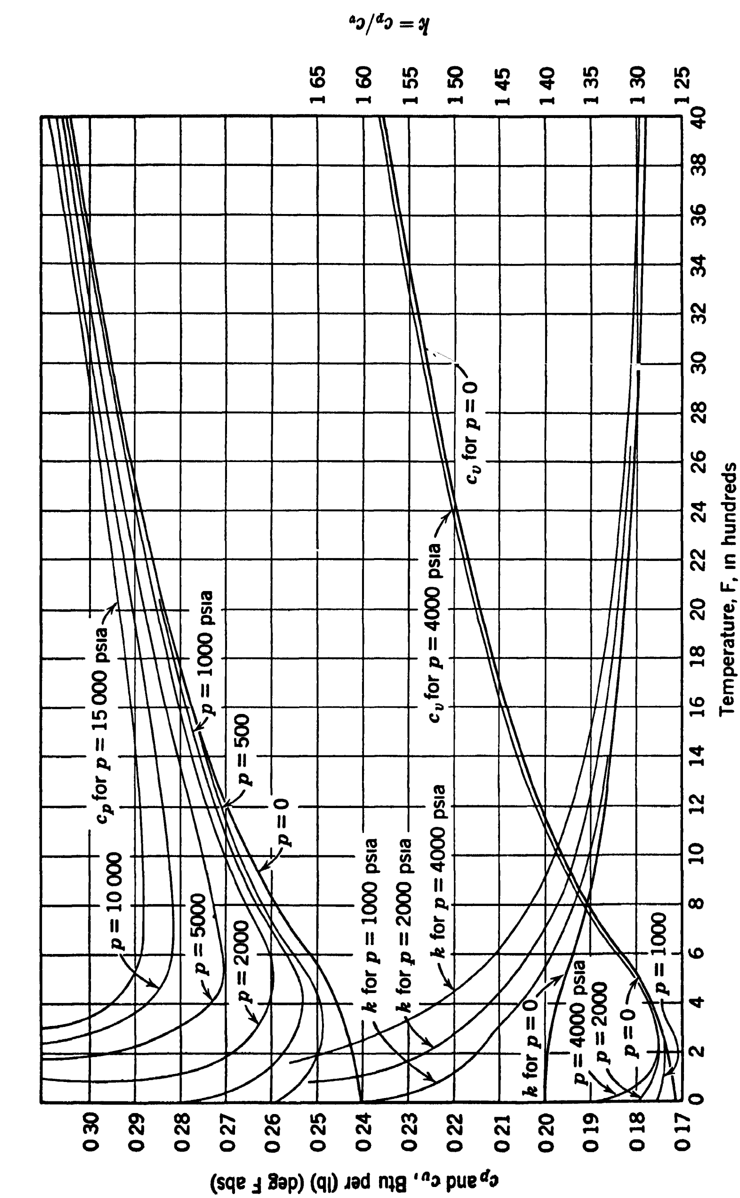


FIG 30 The effect of temperature on c_p, c_v, and k of dry air at various pressures[4]

[4] From F O Ellenwood, N Kulick, and N R Gay, "The Specific Heats of Certain Gases over Wide Ranges of Pressures and Temperatures," *Cornell University Bull* 30, Oct , 1942

necting the two isotherms can be used to determine the relation between dU and dT. Consider the path A_1A_2, for which the energy absorbed by m pounds of gas at constant volume is

$$dQ = mc_v\,dT \tag{60}$$

This expression is a definition of the specific heat of a gas for the particular manner (constant volume) in which the energy is added (see Art. 43).

When a gas is heated at constant volume, the nonflow energy equation $dQ = dU + P\,dV/J$ reduces to $dQ = dU$ since dV is zero. Hence, from equation 60 and $dQ = dU$, the change of internal energy for m pounds is

$$dU = mc_v\,dT \tag{61}$$

Equation 61 is applicable not only to a constant-volume path, such as the one used in its derivation, but to any path; therefore, for a perfect gas, U is a function of temperature only. Integration of equation 61 is

$$U_2 - U_1 = mc_v(T_2 - T_1) \tag{62}$$

Equation 62 may be used to determine the change in internal energy of a perfect gas for *any* change of state for *any* thermodynamic process. Joule's law is correct only for perfect gases, is approximately correct for nearly perfect gases, and is decidedly in error for vapors.

Because of the low coefficients of expansion for liquids and solids, there is little difference in their specific heats at constant volume and at constant pressure. However, with gases, the difference is appreciable.

Specific heat varies widely with temperature but only slightly with pressure as shown by Fig. 30, which illustrates the behavior of dry air. Other gases have similar characteristics which can be expressed in equation form as shown in Table II; these equations are given on a unit mole basis for variable specific heats of gases at constant pressure. To use the equations for 1-pound weight it is necessary to divide by the molecular weight of the gas.

Example. Find the energy required to heat 1 lb of air from 1000 F abs to 2000 F abs at constant pressure, (*a*) using variable specific heat, Table II; (*b*) using constant specific heat ($c_p = 0.24$); and (*c*) using an estimated mean value of c_{pm} from Fig. 30.

Solution. (*a*) $${}_1Q_2 = \frac{m}{\text{mole wt}} \int_{T_1}^{T_2} C_p\,dT$$

$${}_1Q_2 = \frac{1}{28.96} \int_{T_1}^{T_2} (6.3712 + 0.000886176T)\,dT$$

$${}_1Q_2 = 267 \text{ Btu per lb}$$

TABLE II

Specific Heat Equations, Btu per (mole) (deg Fabs)

Gas	Molecular Wt	Molar C_p Equation*	Range, deg F abs	Maximum Error in Per Cent
Air	28.960	$C_p = 6.3712 + 0.000886176T$	720–2000	1.0
O_2	32.000	$C_p = 11.515 - \frac{172}{\sqrt{T}} + \frac{1530}{T}$	540–5000	1.1
N_2	28.020	$C_p = 9.47 - \frac{3.47 \times 10^3}{T} + \frac{1.16 \times 10^6}{T^2}$	540–9000	1.7
H_2	2.016	$C_p = 5.76 + \frac{0.578}{1000} T + \frac{20}{\sqrt{T}}$	540–4000	0.8
CO_2	44.000	$C_p = 16.2 - \frac{6.53 \times 10^3}{T} + \frac{1.41 \times 10^6}{T^2}$	540–6300	0.8
CO	28.000	$C_p = 9.46 - \frac{3.29 \times 10^3}{T} + \frac{1.07 \times 10^6}{T^2}$	540–9000	1.1
H_2O	18.016	$C_p = 19.86 - \frac{597}{\sqrt{T}} + \frac{7500}{T}$	540–5400	1.8
CH_4	16.030	$C_p = 4.52 + 0.00737T$	540–1500	1.2
C_2H_4	28.030	$C_p = 4.23 + 0.01177T$	350–1100	1.5
C_2H_6	30.050	$C_p = 4.01 + 0.01636T$	400–1100	1.5
C_8H_{18}	114.140	$C_p = 7.92 + 0.0601T$	400–1100	4.0 Est.
$C_{12}H_{26}$	170.300	$C_p = 8.68 + 0.0889T$	400–1100	4.0 Est.

From R. L. Sweigert and M. W. Beardsley, "Empirical Specific Heat Equations Based upon Spectroscopic Data," *Georgia School of Technology, Bull.* 2, June, 1938.

* For C_v equations note that $C_p = C_v + R/J = C_v + 1.986$ or $C_v = C_p - 1.986$ for all near-perfect gases on mole basis.

(*b*) Using constant specific heat for air at 0.24 Btu per (lb)(deg F),

$$_1Q_2 = mc_p(T_2 - T_1) = 1 \times 0.24(2000 - 1000)$$

$$_1Q_2 = 240 \text{ Btu per lb}$$

(*c*) The mean specific heat c_{pm} for the given temperature and pressure from Fig. 30 is approximately 0.265; thus

$$_1Q_2 = mc_{pm}(T_2 - T_1)$$
$$= 1 \times 0.265 \times 1000 = 265 \text{ Btu per lb}$$

Note that the results in (*a*) and (*c*) are satisfactory whereas (*b*) illustrates the limitations on the use of constant specific heat for any real gas where a wide range of temperatures is involved. Integrating over a range of temperatures is laborious; consequently reference to a table in which the computations have been made or the use of a mean value (c_{pm} or c_{vm}) is more expedient.

38. Specific Heat Relations. Using ${}_1Q_2 = mc_p(T_2 - T_1)$, equation 59, $U_2 - U_1 = mc_v(T_2 - T_1)$, equation 62, $W = \int_1^2 P\,dV$, equation 28, and $PV = mRT$, equation 37, in ${}_1Q_2 = (U_2 - U_1) + \frac{{}_1W_2}{J}$, equation 27, the following is obtained

$$mc_p(T_2 - T_1) = mc_v(T_2 - T_1) + \frac{mR}{J}(T_2 - T_1) \tag{63}$$

which simplifies to

$$c_p - c_v = \frac{R}{J} \tag{64}$$

Equation 64 shows that c_p and c_v for a particular gas differ by R/J for that gas.

If each term in equation 64 is multiplied by the molecular weight of the gas, the molar specific heat relation results; i.e.,

$$C_p - C_v = \frac{B}{J} = \frac{1545}{778.26} = 1.986 \text{ Btu per (mole) (deg F)} \tag{65}$$

where $C_p = Mc_p$ = molar specific heat of constant pressure, Btu per (mole) (deg F).
$C_v = Mc_v$ = molar specific heat of constant volume, Btu per (mole) (deg F).

Equation 65 is applicable to all perfect gases.

The ratio of c_p to c_v is k where k is a constant for perfect gases and is a constant for real gases within limits (see Fig. 30 and Table I).

From the ratio $c_p/c_v = k$ and equation 64

$$c_p = \frac{kR}{J(k-1)} \quad \text{and} \quad c_v = \frac{R}{J(k-1)} \quad \text{lb basis} \tag{66}$$

From the ratio $C_p/C_v = k$ and equation 65

$$C_p = \frac{kB}{J(k-1)} \quad \text{and} \quad C_v = \frac{B}{J(k-1)} \quad \text{mole basis} \tag{67}$$

When two states of a gas are given in terms of pressure and volume, the difference in internal energy at the two states may be found as follows:

$$PV = mRT$$

$$T_1 = \frac{P_1V_1}{mR} \quad \text{and} \quad T_2 = \frac{P_2V_2}{mR} \tag{68}$$

Substituting these values in $U_2 - U_1 = mc_v(T_2 - T_1)$, equation 62, gives

$$U_2 - U_1 = \frac{c_v}{R}(P_2V_2 - P_1V_1) \tag{69}$$

By substitution of c_v from equation 66, the above equation reduces to

$$U_2 - U_1 = \frac{P_2V_2 - P_1V_1}{J(k-1)} \tag{70}$$

This equation may be used to determine the change in internal energy of a gas for *any* change of state during *any* thermodynamic process.

39. Enthalpy of Perfect Gases. The defining expression for enthalpy, equation 11, is

$$h = u + \frac{Pv}{J} \quad \text{Btu per lb}$$

Hence a change from any state 1 to state 2 will produce an enthalpy change of

$$h_2 - h_1 = (u_2 - u_1) + \frac{P_2v_2 - P_1v_1}{J} \quad \text{Btu per lb} \tag{71}$$

Equation 71 is true for any medium. By application of the perfect gas relations a simpler expression which is true only for a perfect gas can be had. Substituting $u_2 - u_1 = mc_v(T_2 - T_1)$, $P_1V_1 = mRT_1$, and $P_2V_2 = mRT_2$ in equation 71 gives

$$H_2 - H_1 = mc_v(T_2 - T_1) + \frac{mR}{J}(T_2 - T_1) \quad \text{Btu}^4 \tag{72}$$

$$H_2 - H_1 = m\left(c_v + \frac{R}{J}\right)(T_2 - T_1) \tag{73}$$

Since

$$c_p = c_v + \frac{R}{J}$$

$$H_2 - H_1 = mc_p(T_2 - T_1) \quad \text{Btu} \tag{74}$$

If c_p is considered over a wide range of temperature,

$$H_2 - H_1 = m\int_1^2 c_p\,dT \tag{75}$$

The above equation shows that the enthalpy of a perfect gas is a function of temperature only, as is its internal energy.

[4] Note that H represents total quantity and h represents specific quantity.

40. Reversibility. If friction is considered in a broad sense—mechanical friction in mechanical things, turbulent friction in the movement of gases and fluids, flow resistance or "friction" in the movement of heat across a boundary—it can be said that a reversible process is a frictionless process. In reality, there are no frictionless processes and there are no reversible processes. The reversible process, therefore, becomes an ideal against which a real process may be measured. More specifically, a reversible process is one which, after it is accomplished, can be reversed and thereby everything in the process and surroundings can be restored to its original state. Reversible processes cannot involve friction, nor unrestrained expansion of a gas, nor transfer of heat across a finite temperature difference. Since it is quite difficult to evaluate friction effects, it is often much easier to assume reversibility in theoretical considerations and apply a friction factor when needed.

To be reversible the expansion of a gas must take place with no losses due to turbulence or molecular friction. A well-designed nozzle is one of the most efficient mechanical devices because it is capable of transforming the energy of a medium, such as steam, into kinetic energy with an efficiency approaching 97 per cent. Thus flow through a nozzle approaches a reversible process. Use of the kinetic energy generated by the nozzle to return the steam through the nozzle to its original state would be a highly irreversible process since no compressor can approach 100 per cent efficiency.

A reversible transfer of heat requires that heat flow from one medium to another without a temperature difference. Such a condition could be approached but never reached. It is convenient to assume for theoretical considerations that reversible heat transfer can be accomplished.

41. Equation of the Path of a Gas Process. If either the expansion or the compression curve from the pressure-volume diagram of a steam engine, internal-combustion engine, air compressor, or other reciprocating heat engine be transferred point by point to linear coordinates in which log P is plotted against log V, the points so transferred usually fall along a straight line having the equation

$$\log P + n \log V = \log \text{constant} \tag{76}$$

The slope of the line is $-n$, and equation 76 becomes

$$PV^n = \text{a constant} \tag{77}$$

Equation 77 is called the *equation of the path of a gas process.*

Only one straight line can be drawn through two given points on the logarithmic PV diagram; consequently, only one curve can be drawn through the corresponding points on the PV plane. If the values of P

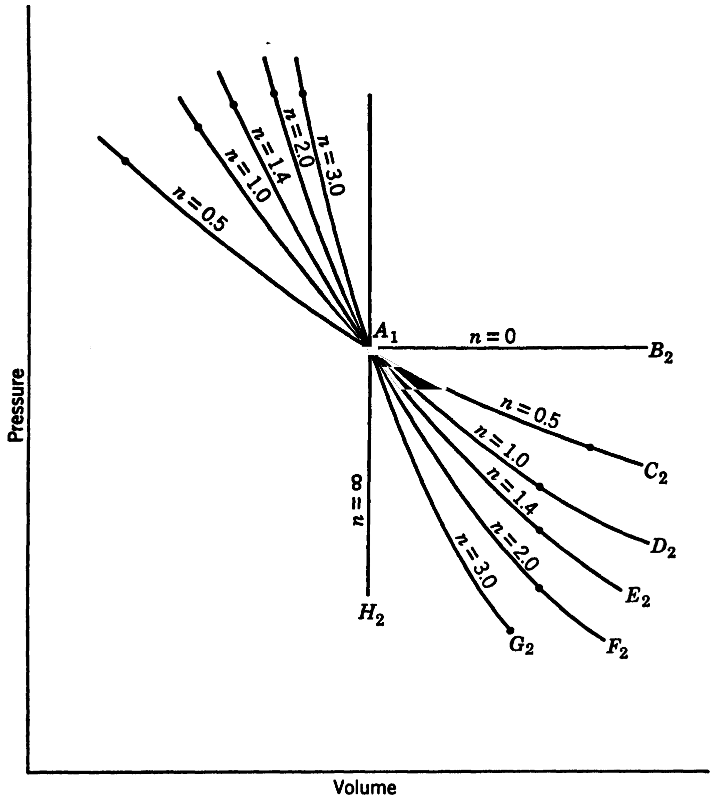


FIG. 31. Paths of gas processes on PV plane.

and V are known at two states of a gas, then

$$PV^n = P_1V_1^n = P_2V_2^n \tag{78}$$

This equation may be solved for n; thus

$$n = \frac{\log P_2 - \log P_1}{\log V_1 - \log V_2} = \frac{\log \dfrac{P_2}{P_1}}{\log \dfrac{V_1}{V_2}} \tag{79}$$

During expansion or compression, n in equations 78 and 79 may have any value between zero and infinity. Figure 31 shows various paths for a process during expansion (or compression) of a gas.

42. Constant-Pressure Process. A nonflow change of state in which the pressure is maintained constant is represented by a horizontal line on the PV plane, A_1B_2 in Fig. 31. For this process, equation 38

reduces to

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{80}$$

The work done, from equation 28, is

$${}_1W_2 = \int_1^2 P\,dV = P(V_2 - V_1) \quad \text{ft-lb} \tag{81}$$

The change of internal energy, from equations 70 and 60, is

$$U_2 - U_1 = \frac{P(V_2 - V_1)}{J(k-1)} = mc_v(T_2 - T_1) \quad \text{Btu} \tag{82}$$

using an average value for c_v. Therefore, the heat absorbed during a constant-pressure process is, by substitution in equation 27,

$${}_1Q_2 = mc_v(T_2 - T_1) + \frac{P(V_2 - V_1)}{J} \quad \text{Btu} \tag{83}$$

Equation 83 shows that, when heat energy is absorbed by a gas at constant pressure, part of the energy is stored as internal energy and part produces work.

The heat absorbed during a constant-pressure process, using an average value for c_p, is given by

$${}_1Q_2 = \int_1^2 mc_p\,dT \tag{84}$$

$${}_1Q_2 = mc_p(T_2 - T_1) \tag{85}$$

From equation 74 the change of enthalpy is

$$H_2 - H_1 = {}_1Q_2 = mc_p(T_2 - T_1) \tag{86}$$

43. Constant-Volume Process. A nonflow change of state in which the volume is maintained constant is represented by a vertical line on the PV plane, A_1H_2 in Fig. 31. For this process, equation 35 reduces to $P_1/P_2 = T_1/T_2$. Under the condition of constant volume no work can be performed,

$${}_1W_2 = \int_1^2 P\,dV = 0 \tag{87}$$

Therefore, from the simple energy equation, $Q = (U_2 - U_1) + W/J$, the quantity of heat added is

$${}_1Q_2 = U_2 - U_1 \tag{88}$$

From equation 70, and since $V_1 = V_2$,

$$_1Q_2 = \frac{V(P_2 - P_1)}{J(k - 1)} \tag{89}$$

Using a mean value of c_v,

$$_1Q_2 = mc_v(T_2 - T_1) \tag{90}$$

Since enthalpy has meaning only for a flow process, the result has no significance when $H_2 - H_1 = mc_p(T_2 - T_1)$ is evaluated for a constant-volume process.

Example. A tank having a volume of 25 cu ft contains air at a pressure of 14.5 psia and a temperature of 70 F. How many Btu will be required to heat the air to a temperature of 150 F? Assume constant mean specific heat, $c_v = 0.171$.

Solution. $P_1V_1 = mRT_1$

$$14.5 \times 144 \times 25 = m \times 53.35(70 + 459.7)$$

$$m = 1.85 \text{ lb of air}$$

$$_1Q_2 = mc_v(T_2 - T_1)$$

$$_1Q_2 = 1.85 \times 0.171(609.7 - 529.7) = 25.3 \text{ Btu}$$

44. Isothermal Process. A special case of practical importance is that of an expansion, or a compression, during which the temperature of the gas remains constant. Such a change of state is called *isothermal.* When a gas is expanding isothermally, heat is added to maintain a constant temperature; the gas, therefore, must be in contact with an external source of heat. When compressed isothermally a gas must be in contact with a receiver which can take heat from it; for example, the water jacket of an air compressor receives part of the heat of compression. If T is made constant, equation 37, $PV = mRT$, gives

$$PV = \text{a constant} \tag{91}$$

and

$$P_1V_1 = P_2V_2 = P_3V_3 = \cdots = P_nV_n \tag{92}$$

The above statement is the equation of the isotherm A_1D_2, Fig. 31. The isothermal process shown on the PV plane is an equilateral hyperbola, PV^n = a constant, $n = 1$.

From equation 28,

$$_1W_2 = \int_1^2 P\,dV$$

and equation 92,

$$P = P_1V_1/V$$

it can be seen that

$$ {}_1W_2 = P_1 V_1 \int_1^2 \frac{dV}{V} \tag{93} $$

Integrating and substituting limits,

$$ {}_1W_2 = P_1 V_1 (\log_e V_2 - \log_e V_1) $$

or

$$ {}_1W_2 = P_1 V_1 \log_e \frac{V_2}{V_1} \quad \text{ft-lb} \tag{94} $$

But since

$$ PV = P_1 V_1 \quad \text{and} \quad PV = mRT $$

then

$$ {}_1W_2 = mRT \log_e \frac{V_2}{V_1} \quad \text{ft-lb} \tag{95} $$

where T is the temperature (deg F abs) at which the process takes place.

During the isothermal expansion, or compression, of a perfect gas there is no change of temperature; hence there is no change of internal energy; from equation 61, $dU = mc_v\,dT$,

$$ U_2 - U_1 = 0 \tag{96} $$

From equation 75,

$$ H_2 - H_1 = \int_1^2 mc_p\,dT $$

the change of enthalpy for an isothermal process is

$$ H_2 - H_1 = 0 \tag{97} $$

Hence, during the isothermal expansion of a gas, an amount of heat must be absorbed equivalent to the work done; from equation 27,

$$ {}_1Q_2 = (U_2 - U_1) + \frac{{}_1W_2}{J} \tag{98} $$

$$ {}_1Q_2 = \frac{{}_1W_2}{J} $$

Substituting equation 95 in the above equation gives

$$ {}_1Q_2 = \frac{mRT}{J} \log_e \frac{V_2}{V_1} \quad \text{Btu} \tag{99} $$

Also from equation 94

$$ {}_1Q_2 = \frac{P_1 V_1}{J} \log_e \frac{V_2}{V_1} \quad \text{Btu} \tag{100} $$

Thus during an isothermal process the change of internal energy and enthalpy are both zero, and any heat added appears as work. During isothermal compression the gas must reject heat equivalent to the work done on it.

Example. Three cubic feet of air under a pressure of 80 psia are expanded isothermally to a pressure of 20 psia. Find (*a*) the work done during this expansion and (*b*) the heat added in Btu.

Solution. (*a*) From equation 92

$$V_2 = \frac{3 \times 80 \times 144}{20 \times 144} = 12 \text{ cu ft}$$

From equation 94

$${}_1W_2 = 144 \times 80 \times 3 \log_e \frac{12}{3}$$

$${}_1W_2 = 47{,}970 \text{ ft-lb}$$

(*b*)

$${}_1Q_2 = \frac{{}_1W_2}{J}$$

$${}_1Q_2 = \frac{47{,}970}{778.26} = 61.7 \text{ Btu}$$

45. Entropy. Clausius defined this thermodynamic property thus, "If in any reversible process an element of heat taken in (positive or negative) be divided by the absolute temperature at which it is taken in, the result is equal to the corresponding change in *entropy*."

Stated mathematically, for a reversible process,

$$ds = \frac{dQ}{T} \quad \text{or} \quad s_2 - s_1 = \int_1^2 \frac{dQ}{T} \tag{101}$$

where $(s_2 - s_1)$ represents the change of entropy and the other symbols have their previous meanings. From equation 101, for a reversible process,

$$dQ = T\,ds$$

or

$${}_1Q_2 = \int_1^2 T\,ds \tag{102}$$

If the temperature T is constant during the reversible heat transfer,

$${}_1Q_2 = T(s_2 - s_1) \tag{103}$$

In words, equation 103 states that the heat supplied, or rejected, in the process from state point 1 to state point 2 is equal to the absolute temperature T (deg F abs) multiplied by the entropy change. Figure 32

illustrates equation 103 and shows one use of the concept of entropy, i.e., that of a coordinate.

The change of entropy $(s_b - s_a)$ may be seen in Fig. 32 as $(s_b - s_a) = {}_1Q_2/T$ for this constant-temperature reversible process. When drawn to scale, the area a–1–2–b–a represents the heat transferred. Usually T is not constant as at point x and ${}_0Q_1 = \int_0^1 T\,ds$.

Entropy is thus a useful coordinate or parameter for graphically and analytically dealing with energy quantities. Since the change in entropy is more often desired than absolute values, it is of little impor-

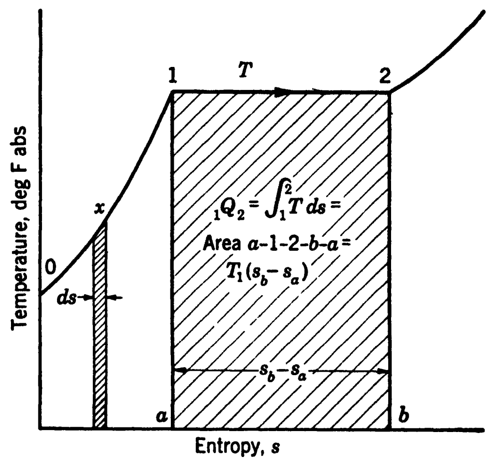


FIG. 32. Temperature-entropy diagram.

tance where the zero value of entropy, s_0 in Fig. 32, is established. Arbitrarily, in most tables, 32 F is used as zero entropy for steam, mercury vapor, and carbon dioxide, −40 F for refrigerants, and −360 F or 100 F abs for air.

Entropy is a property of any medium since its value is invariant for a given medium at a given state point regardless of how that state point was reached. Therefore, entropy values for a given medium can be calculated and compiled in table form. Entropy is tabulated in engineering units as Btu per pound weight, per degree F absolute.

Entropy can be used with many other properties (T, P, V, U, and H) for the graphical representation of state changes. The most common are the temperature-entropy and enthalpy-entropy diagrams. The latter is called a Mollier diagram, Fig. 39, and is of great value in dealing with media whose normal working range is such that they pass

from a liquid to a vapor state and vice versa—steam, ammonia, and Freon—Art. 65.

For a reversible adiabatic process the entropy is constant and the process is said to be *isentropic*. An isentropic expansion, or compression, serves as a standard of perfection since all heat transfer losses and friction effects have been assumed to be zero.

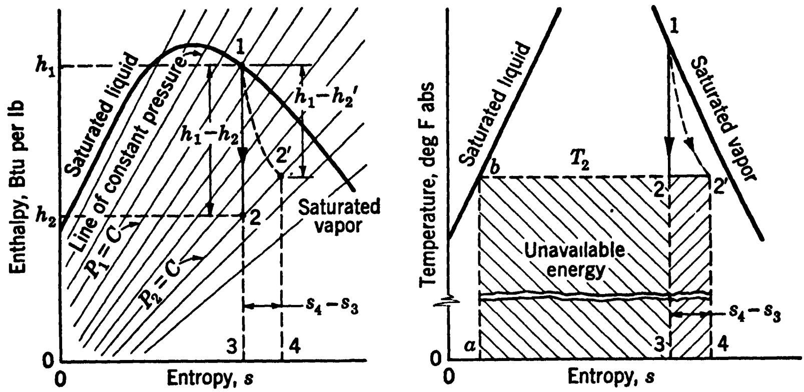


FIG. 33. Reversible and irreversible expansion.

Figure 33 illustrates an isentropic expansion as occurring between points 1 and 2 on the enthalpy-entropy plane and also on the temperature-entropy plane. Such an expansion, which illustrates steady flow through a nozzle, shows that the maximum change in enthalpy, $h_1 - h_2$, has occurred, and by the steady-flow equation this energy must appear in some other form such as kinetic energy or work. If the process is irreversible from 1 to 2′, the entropy increases as shown by $(s_4 - s_3)$. The change of enthalpy, $h_1 - h_2'$, by the irreversible path is less than by the reversible path; consequently, less work can be obtained by such an expansion. If on the Ts plane, Fig. 33, T_2 is the lowest temperature (sink) at which the energy is usable, area on this plane below T_2 represents unavailable energy. Due to the irreversibility of the process this area is increased by the area $T_2(s_4 - s_3)$ for each pound of medium.

The entropy change can be calculated by equation 101 whether the process is reversible or irreversible since two reversible processes can always be assumed which will replace the one irreversible operation. In Fig. 33 the change in entropy from 1 to 2′ by the irreversible path is equal to the change in entropy by the two reversible paths, 1 to 2 (reversible expansion) and 2 to 2′ (reversible transfer of heat).

When values are substituted for dQ in equation 101, $ds = dQ/T$, equations for entropy change are obtained for various processes. From ${}_1Q_2 = U_2 - U_1 + \frac{{}_1W_2}{J}$, for a reversible process,

$$dQ = dU + \frac{P\,dv}{J} \tag{104}$$

Since

$$dQ = T\,ds$$

$$ds = \frac{dU}{T} + \frac{P\,dv}{JT}$$

But

$$dU = c_v\,dT \quad \text{and} \quad P = \frac{RT}{v} \quad (\text{from } Pv = RT)$$

Therefore

$$ds = \frac{c_v\,dT}{T} + \frac{R\,dv}{Jv}$$

Integrating,

$$s_2 - s_1 = \int_{T_1}^{T_2} c_v \frac{dT}{T} + \frac{R}{J} \log_e \frac{v_2}{v_1} \tag{105}$$

Since

$$c_v = c_p - \frac{R}{J}$$

$$ds = c_p \frac{dT}{T} - \frac{R\,dT}{JT} + \frac{R\,dv}{Jv}$$

Also, since

$$P\,dv + v\,dP = R\,dT \quad \text{or} \quad \frac{dv}{v} + \frac{dP}{P} = \frac{dT}{T}$$

$$ds = c_p \frac{dT}{T} - \frac{R\,dP}{JP}$$

Integrating,

$$s_2 - s_1 = \int_{T_1}^{T_2} c_p \frac{dT}{T} - \frac{R}{J} \log_e \frac{P_2}{P_1} \tag{106}$$

Equations 105 and 106 can be used for calculating the change of entropy between any two states

where s = entropy, Btu per (lb) (deg F abs).
R = individual gas constant, ft-lb_f per (lb_m) (deg F abs).
T = temperature, deg F abs.

v = specific volume, cu ft per lb.
P = pressure, psfa.
c_v = specific heat of constant volume, Btu per (lb) (deg F).
c_p = specific heat of constant pressure, Btu per (lb) (deg F).

A more general equation is obtained when molar C_v and C_p are substituted, respectively, for c_v and c_p on a pound weight basis and B/J = 1.986 replaces R/J, B/J being a constant for any perfect gas or a mixture of perfect gases. If, in addition, the temperature range is small so that C_v and C_p are considered constant, then, on a mole basis, equations 105 and 106 are, respectively,

$$s_2 - s_1 = C_v \log_e \frac{T_2}{T_1} + \frac{B}{J} \log_e \frac{V_2}{V_1} \quad \text{Btu per (mole) (deg F abs)} \qquad (107)$$

or

$$s_2 - s_1 = C_p \log_e \frac{T_2}{T_1} - \frac{B}{J} \log_e \frac{P_2}{P_1} \quad \text{Btu per (mole) (deg F abs)} \qquad (108)$$

and, on a pound weight basis, with constant specific heat,

$$s_2 - s_1 = c_v \log_e \frac{T_2}{T_1} + \frac{R}{J} \log_e \frac{v_2}{v_1} \quad \text{Btu per (lb) (deg F abs)} \qquad (109)$$

$$s_2 - s_1 = c_p \log_e \frac{T_2}{T_1} - \frac{R}{J} \log_e \frac{P_2}{P_1} \quad \text{Btu per (lb) (deg F abs)} \qquad (110)$$

For a *constant-volume process*, equation 109 reduces to

$$s_2 - s_1 = c_v \log_e \frac{T_2}{T_1} \quad \text{or} \quad c_v \log_e \frac{P_2}{P_1} \quad \text{Btu per (lb) (deg F abs)} \qquad (111)$$

For a *constant-pressure* process, equation 110 reduces to

$$s_2 - s_1 = c_p \log_e \frac{T_2}{T_1} \quad \text{or} \quad c_p \log_e \frac{v_2}{v_1} \quad \text{Btu per (lb) (deg F abs)} \qquad (112)$$

For an *isothermal process*, equation 105 reduces to

$$s_2 - s_1 = \frac{R}{J} \log_e \frac{v_2}{v_1} \quad \text{or} \quad \frac{R}{J} \log_e \frac{P_1}{P_2} \quad \text{Btu per (lb) (deg F abs)} \qquad (113)$$

46. Adiabatic Processes. An adiabatic process is one during which no heat is transferred to or from the working substance. A reversible adiabatic process is called an *isentropic* (*constant-entropy*) *process.* By definition, no heat energy enters or leaves the working substance during an isentropic process; i.e., ${}_1Q_2 = 0$, and the simple energy equation 25 becomes

$$(U_2 - U_1) + \frac{{}_1W_2}{J} = 0$$

or

$$mc_v\,dT + \frac{P\,dV}{J} = 0 \qquad (114)$$

During the isentropic process, the pressure, volume, and temperature vary; differentiating the characteristic equation,

$$PV = mRT$$

$$P\,dV + V\,dP = mR\,dT$$

and

$$dT = \frac{P\,dV + V\,dP}{mR}$$

This value of dT substituted in equation 114 gives

$$\frac{c_vP\,dV}{R} + \frac{c_vV\,dP}{R} + \frac{P\,dV}{J} = 0$$

Simplifying gives

$$P\,dV\left(c_v + \frac{R}{J}\right) + c_vV\,dP = 0$$

From equation 64

$$c_v + \frac{R}{J} = c_p$$

hence

$$c_pP\,dV + c_vV\,dP = 0$$

Separating variables and noting that $c_p/c_v = k$,

$$\frac{dP}{P} + k\frac{dV}{V} = 0$$

This equation integrates to

$$\log_e P + k \log_e V = \log_e \text{constant}$$

Taking the antilogarithm of the above equation,

$$PV^k = \text{a constant} \qquad (115)$$

Therefore, for an isentropic process, if the values of P and V are known at two states of the gas,

$$PV^k = P_1V_1^k = P_2V_2^k = C \qquad (116)$$

Combining equations 35 and 116 gives

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1} = \left(\frac{P_1}{P_2}\right)^{(k-1)/k} \qquad (117)$$

Equations 116 and 117 will be useful for isentropic calculations and in deriving special forms of other equations. These equations are applicable only to perfect gases and are not valid if the specific heat varies. The isentropic process is defined for either a gas or a vapor by the relation ${}_1Q_2 = 0$ (reversibility assumed); hence, from the simple energy equation, $(U_2 - U_1) = -{}_1W_2$.

The amount of *work done* during the isentropic process, by equation 28, is

$${}_1W_2 = \int_1^2 P\,dV$$

Since the isentropic process is such that $PV^k = P_1V_1{}^k$, by equation 116, P at *any* point is $P_1V_1{}^k/V^k$, P_1 and V_1 being the pressure and volume at the initial state. Then, for expansion from V_1 to V_2,

$${}_1W_2 = P_1V_1{}^k \int_1^2 \frac{dV}{V^k}$$

which gives by integration

$${}_1W_2 = \frac{P_1V_1{}^k(V_2{}^{1-k} - V_1{}^{1-k})}{1 - k}$$

$${}_1W_2 = \frac{P_2V_2 - P_1V_1}{1 - k} \quad \text{ft-lb} \qquad (118)$$

The work done during an *isentropic process* may be determined by equation 118. By definition, the *transferred heat* is zero. Thus, from equation 27, the increase of the *internal energy* is equal to the work done on a gas during an isentropic process. Similarly, when a gas does work during an isentropic process, it loses internal energy equal to the work done. For a perfect gas, the change of *enthalpy* is found as explained in Art. 39, and, by means of equation 74, $H_2 - H_1 = mc_p(T_2 - T_1)$. Since ${}_1Q_2 = 0$, from equation 101, $ds = 0$; by integration, $s_2 - s_1 = 0$. Therefore during a reversible adiabatic process the entropy is constant; hence the process is isentropic.

In actual heat engines the action is never strictly adiabatic because there are always some exchanges of heat between the working substance and the metallic surfaces of the cylinder and piston. Rapid expansion or compression approaches being adiabatic by giving little time for heat transfer. There are many cases in actual practice where *irreversible adiabatic expansion* is encountered, in which there is a drop from a condition of high pressure to one of lower pressure without any useful work being done; for example, when air flows through an orifice from a high-pressure tank into the atmosphere or when a fluid passes

through a pressure-reducing valve or a partly open valve. Such a process is called *throttling* or *wiredrawing*. The total amount of energy in the fluid remains constant, $H_1 = H_2$, but its availability for doing work has decreased, as illustrated in Fig. 33.

For a perfect gas, equation 74 states that $(H_2 - H_1) = mc_p(T_2 - T_1)$, and, since $H_1 = H_2$, it follows that $T_1 = T_2$; i.e., the initial and final temperatures are equal in the throttling process for a perfect gas. Since entropy is a point function, the change of entropy between any two states can be computed for any path or series of paths connecting the two points. Hence the change of entropy in the throttling process for a perfect gas is the same as for the reversible isothermal process,

$$s_2 - s_1 = \frac{R}{J} \log_e \frac{v_2}{v_1} \quad \text{Btu per lb} \tag{119}$$

For throttling processes of *perfect gases*, $P_1 V_1 = P_2 V_2$, since the initial and final temperatures are the same. However, in a constant-enthalpy process, *actual gases* do show an appreciable change of temperature for large pressure drops.

The principles of the irreversible steady-flow adiabatic (throttling) process are generally used in heat-power practice when liquids and vapors are involved.

Example. Three cubic feet of air entered an ideal air compressor on the suction stroke at atmospheric pressure (14.5 psia) and were compressed adiabatically to a pressure of 87 psia. (*a*) Find the work done during compression. (*b*) What was the final temperature if the initial temperature was 68 F?

Solution. (*a*) This was an isentropic process; hence, from equation 116,

$$V_2 = V_1 \left(\frac{P_1}{P_2}\right)^{1/k} \qquad k = 1.4 \text{ for air (from Table I)}$$

$$V_2 = 3 \left(\frac{14.5}{87}\right)^{1/1.4} = 0.834 \text{ cu ft}$$

Substituting in equation 118 gives

$$_1W_2 = \frac{144[(87 \times 0.834) - (14.5 \times 3)]}{-0.40}$$

$$= -10{,}461 \text{ ft-lb}$$

(*b*) Rearranging and substituting in equation 117 gives

$$T_2 = 527.7 \left(\frac{87}{14.5}\right)^{0.4/1.4} = 880.4 \text{ deg F abs}$$

$$t_2 = 420.7 \text{ F}$$

47. Polytropic Processes. As indicated in the preceding discussions, the value of n in equation 77, PV^n = a constant, will determine whether

heat has been absorbed or rejected or remains constant and whether the temperature rises, falls, or does not change during any given expansion or compression. Equation 78 is the equation of a group of hyperbolas (see Fig. 31), and by assigning special values to n the PV^n relationships of the preceding discussions may be duplicated.

If $n = 0$, PV^0 = constant or P = constant; constant-pressure process.
If $n = 1$, PV = constant or T = constant; isothermal process.
If $n = k$, PV^k = constant or $dQ = 0$; adiabatic process.
If $n = \infty$, PV^∞ = constant or V = constant; constant-volume process.

A polytropic process is one during which the change of state for any working substance conforms to the relation

$$PV^n = \text{a constant} \tag{120}$$

where n may be assigned *any* value (depending upon the process) but does not vary when so assigned; the four cases above are only specialized forms of this general equation. The property relations and energy changes involved during this process may be determined in a manner similar to that employed for the reversible adiabatic (isentropic) process, Art. 46. Hence

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{(n-1)} = \left(\frac{P_1}{P_2}\right)^{(n-1)/n} \tag{121}$$

$$_1W_2 = \frac{P_2V_2 - P_1V_1}{1 - n} \quad \text{ft-lb} \tag{122}$$

$$U_2 - U_1 = mc_v(T_2 - T_1) \quad \text{Btu} \tag{123}$$

$$_1Q_2 = mc_v(T_2 - T_1) + \frac{P_2V_2 - P_1V_1}{J(1 - n)}$$

But $PV = mRT$; therefore

$$_1Q_2 = mc_v(T_2 - T_1) + \frac{mR(T_2 - T_1)}{J(1 - n)}$$

and, from equation 66,

$$R = c_vJ(k - 1)$$

therefore,

$$_1Q_2 = mc_v\left(\frac{k - n}{1 - n}\right)(T_2 - T_1) \quad \text{Btu} \tag{124}$$

Equation 124 is of the general form

$$_1Q_2 = mc_n(T_2 - T_1) \tag{125}$$

where

$$c_n = c_v\left(\frac{k - n}{1 - n}\right) \tag{126}$$

is the specific heat of a polytropic process or the ***polytropic specific heat.*** The value of c_n may vary infinitely because n may vary infinitely. The value of c_n will be negative when $k > n > 1$; this negative specific heat indicates that heat is rejected by the working substance although the temperature has increased or that heat is absorbed even though the temperature has decreased.

For the polytropic process of a perfect gas, the change of enthalpy is found by means of equation 74

$$H_2 - H_1 = mc_p(T_2 - T_1) \tag{127}$$

For the reversible polytropic process the change of entropy is dQ/T, where $dQ = mc_n\,dT$, from equation 125. If c_n is constant

$$s_2 - s_1 = mc_n\int_1^2 \frac{dT}{T} = mc_n \log_e \frac{T_2}{T_1} \tag{128}$$

where c_n is determined by equation 126. This equation should be used only for perfect gases, or nearly perfect gases, since c_n was assumed constant; it is applicable only for a limited range of temperature, approximately 300 F, unless c_v and k in equation 126 have been determined as the average values for the given temperature range.

Example. Two pounds of oxygen are compressed polytropically from a pressure of 14 psia and 70 F to 98 psia. The gas constant R is 48.30 (Table I), c_p is 0.217, and the compression is according to $PV^{1.3} = C$. Determine (*a*) the initial volume, (*b*) the final temperature, (*c*) the work done, (*d*) the change of internal energy, (*e*) the heat transferred, (*f*) the change of enthalpy, and (*g*) the change of entropy.

Solution.

(*a*) $$V_1 = \frac{mRT_1}{P_1}, \text{ from equation 37}$$

$$V_1 = \frac{2 \times 48.30(70 + 459.7)}{144 \times 14} = 25.4 \text{ cu ft}$$

(*b*) $$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{(n-1)/n}, \text{ equation 121}$$

$$T_2 = (70 + 459.7)\left(\frac{98}{14}\right)^{0.3/1.3} = 529.7 \times 1.567 = 830 \text{ F abs}$$

$$t_2 = 370.3 \text{ F}$$

(*c*) $$V_2 = \frac{mRT_2}{P_2} = \frac{2 \times 48.30 \times 830}{98 \times 144} = 5.68 \text{ cu ft}$$

$$ {}_1W_2 = \frac{P_2V_2 - P_1V_1}{1 - n}, \quad \text{equation 122} $$

$$ {}_1W_2 = \frac{144[(98 \times 5.68) - (14 \times 25.4)]}{(1 - 1.3)} = -96{,}480 \text{ ft-lb} = -123.9 \text{ Btu} $$

(*d*)
$$ c_v = c_p - \frac{R}{J}, \quad \text{equation 64} $$

$$ c_v = 0.217 - \frac{48.28}{778.26} = 0.217 - 0.062 = 0.155 $$

$$ U_2 - U_1 = mc_v(T_2 - T_1) = 2 \times 0.155(830 - 529.7) = 93.7 \text{ Btu} $$

(*e*)
$$ k = c_p \div c_v = 0.217 \div 0.155 = 1.40 $$

$$ c_n = c_v\left(\frac{k - n}{1 - n}\right) = 0.155\left(\frac{1.40 - 1.30}{1 - 1.30}\right) = -0.052 $$

$$ {}_1Q_2 = mc_n(T_2 - T_1) = 2(-0.052)(830 - 529.7) = -31.2 \text{ Btu} $$

(*f*)
$$ H_2 - H_1 = mc_p(T_2 - T_1), \quad \text{equation 127} $$
$$ = 2 \times 0.217(830 - 529.7) = 131 \text{ Btu} $$

(*g*)
$$ s_2 - s_1 = mc_n \log_e \frac{T_2}{T_1} = 2(-0.052)\log_e \frac{830}{529.7} = -0.0467 \text{ Btu per deg F abs} $$

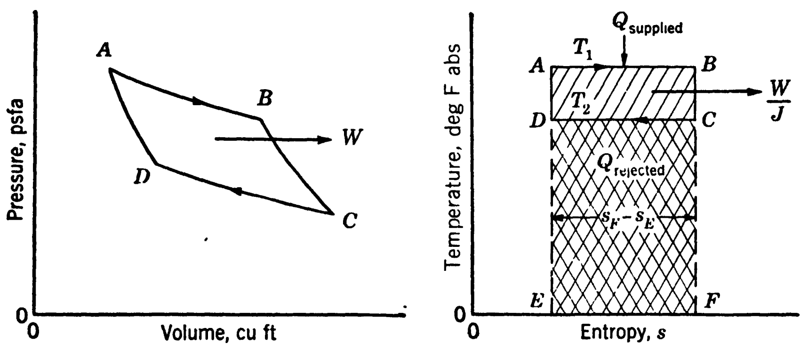


FIG. 34. Carnot cycle.

48. The Carnot Cycle. In 1824 Sadi Carnot published a paper entitled "Reflections on the Motive Power of Heat" in which he stated the concept of reversibility and showed that it was essential to maximum efficiency. Carnot proposed a reversible cycle (illustrated in Fig. 34) in which heat was added to a medium within a cylinder from a reservoir of infinite capacity. All the energy was delivered at temperature T_1 and was transferred to the cylinder contents at the same temperature; thus reversible transfer of heat was assumed. This process provided an

isothermal expansion from A to B, Fig. 34. At point B the heat source was removed and expansion continued isentropically to point C. At this point the piston was reversed as energy was removed from the gases at the temperature T_2 to a sink of infinite capacity, also at temperature T_2. This process was continued to point D at which the sink was disconnected and isentropic compression completed the cycle to point A.

The net work done in this reversible cycle is shown in Fig. 34 as area A–B–C–D–A on the PV plane. The energy supplied is shown on the Ts plane as E–A–B–F–E and the energy discarded to the sink is shown as area E–D–C–F–E. The net difference is the heat equivalent of the work done and is shown as area A-B-C-D-A.

The efficiency is

$$e_c = \frac{Q_{supplied} - Q_{rejected}}{Q_{supplied}} = \frac{W/J}{Q_{supplied}}$$

$$e_c = \frac{T_1(s_F - s_E) - T_2(s_F - s_E)}{T_1(s_F - s_E)}$$

$$= \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1} \qquad (129)$$

From equation 129 it is evident that for maximum efficiency the energy should be delivered to the cycle at the highest possible temperature T_1 and the unused portion rejected to the sink at the lowest possible temperature T_2.

The Carnot cycle provides an illustration of available and unavailable energy. The available energy in this case is shown as area A–B–C–D–A. The unavailable energy is shown as E-D-C-F-E and is equal to the product of the sink temperature and the change in entropy, $T_2(s_F - s_E)$. Unavailable energy represents that part of the total energy which is not usable and must be discarded as dictated by T_2, the lowest temperature available to the cycle.

PROBLEMS

1. A centrifugal pump receives 60 F water at a pressure of 10 psig through a 6-in. pipe and delivers 40 cu ft per min through a 4-in. discharge line at a pressure of 100 psig. If gages are at the center line of the pump, find (*a*) the work done per minute on the water and (*b*) the horsepower required to drive the pump, neglecting mechanical losses.

2. Water having a specific weight of 62.4 lb per cu ft flows downward in a closed flume in which it drops 96 ft vertically and discharges through a 3-in. diameter nozzle to drive a water wheel open to the atmosphere. Neglecting friction and inlet velocity, determine (*a*) the nozzle exit velocity; (*b*) the kinetic energy of flow; and (*c*) the horsepower available.

3. In a nonflow system consisting of a cylinder and frictionless piston, 2000 Btu are added to the medium. If the internal energy does not change, determine the work that will be done.

4. Calculate the specific volume (1 pound) of air at 60 F and 14.696 psia.
13.1 cu ft per lb, *Ans.*

5. Calculate the pound mole or specific mole volume of air at 60 F and 14.696 psia.

6. Ten pounds of hydrogen are contained in a tank at 2000 psia and 60 F. What volume would this hydrogen occupy at 14.696 psia and 90 F?

7. One pound of air in a perfectly insulated cylinder forces the piston a distance of 1 ft on expanding against an average force of 5000 lb. Assuming no friction, find the change in internal energy.

8. A perfectly insulated vessel is divided into two compartments, one holding 1 lb of H_2 at 200 psia and 30 F and the other holding 21 lb of N_2 at 100 psia and 30 F. If the partition is removed, find the final pressure. 125 psia, *Ans.*

9. A perfectly insulated vessel is divided into two compartments, one holding 2 cu ft of air at 50 psia and 60 F and the other holding 3 cu ft of air at 20 psia and 100 F. If the partition is removed, find the pressure and temperature of the mixture. Assume constant specific heat.

10. Eighty pounds of hydrogen contained in a balloon are in equilibrium with the surrounding air at 70 F and 14.696 psia. Find the net lifting capacity if the balloon cloth and rig weigh 43 lb?

11. Ten thousand cubic feet of helium at 14 psia and 60 F are contained in a bag weighing 100 lb. Find the net lifting capacity of the bag when in equilibrium with air. Mole weight of helium = 4.0.

12. A constant-volume air tank of 100 cu ft capacity contains air at 60 F and 80 psia. If the temperature is raised to 120 F, find (*a*) the work done, (*b*) the heat added if c_v is assumed constant, and (*c*) the change in internal energy.

13. A constant-volume tank contains 1 lb of evaporated gasoline C_8H_{18} at 60 F. How many Btu must be supplied to raise its temperature to 360 F, assuming variable specific heat? (See Table II.)

14. In a nonflow constant-pressure process 1 lb of CO_2 is contained in a cylinder at 50 F and 14.696 psia. Heat is added until the volume is doubled as the piston moves. Determine (*a*) the work done, ft-lb; (*b*) the change in temperature, deg F; (*c*) the change in internal energy, Btu; and (*d*) the change in entropy. Use constant specific heats from Table I.

15. In a constant-temperature process 0.5 lb of air is expanded in a cylinder from 100 psia and 212 F to twice its original volume. Determine (*a*) the work done, ft-lb; (*b*) the change in internal energy, Btu; (*c*) the heat supplied; and (*d*) the change in entropy. (*a*) 12,410 ft-lb, (*b*) 0, (*c*) 15.95 Btu, (*d*) 0.0474, *Ans.*

16. In an isentropic nonflow process 0.1 lb of air at 500 psia and 200 F is expanded in a cylinder until the pressure is 14.696 psia. Determine (*a*) the final volume; (*b*) work done, ft-lb; (*c*) the final temperature, deg F abs; (*d*) the change in internal energy; and (*e*) the change in entropy.

17. In a reversible polytropic process 1 cu ft of air at 60 F and 14.696 psia is compressed according to the relation $PV^{1.3} = C$ until the volume is 0.20 cu ft (*a*) Sketch the process on *Ts* and *PV* planes; determine (*b*) the final pressure; (*c*) the work done, ft-lb; (*d*) the final temperature; (*e*) the heat rejected, Btu; and (*f*) the change in internal energy.

18. A tank holds a mixture of 50 lb of oxygen and 50 lb of hydrogen at 70 F and a total pressure of 100 psia. (*a*) How many moles of each gas are in the tank? (*b*)

What is the partial pressure of each gas? (*c*) If the gases were segregated what volume would each occupy in cubic feet?

19. Find the energy required to raise the temperature of 1 lb of air at constant pressure of 14.696 psia from 600 F to 1500 F, assuming (*a*) a constant specific heat of 0.24 Btu per (lb)(deg F); (*b*) a mean specific heat from Fig. 30; and (*c*) by integration from Table II. (*a*) 216 Btu, (*b*) 238 Btu, (*c*) 239.3 Btu, *Ans.*

20. An engine working on the Carnot cycle has 3000 Btu per min supplied at a temperature of 600 F and rejects heat to a sink at a temperature of 80 F. Calculate (*a*) the theoretical horsepower developed and (*b*) the heat rejected, Btu.

21. A Carnot cycle operating between a source at 540 F and a sink at 40 F delivers 100 hp. What quantities of heat must be absorbed and rejected in Btu per minute? Illustrate on the *Ts* plane.

22. A reversed Carnot cycle requires 3 hp and extracts energy from a lake to heat a house. If the house is kept at 70 F and requires 2000 Btu per minute, what is the temperature of the lake? 36 F, *Ans.*

23. Container *A* which is closed, rigid, and insulated contains 3 cu ft of air at 600 psia and 100 F. Container *A* is connected by means of a valve with container *B* which is also closed, rigid, and insulated and contains air at 20 psia and 40 F. When the connecting valve is opened, the resulting equilibrium temperature is 60 F. What is the volume of compartment *B*?

24. The specific heat of a gas at constant pressure is 0.154 and at constant volume is 0.123. If 5 lb of this gas are at a pressure of 50 psia and occupy a volume of 9 cu ft, what is the temperature, deg F?

25. Air in steady flow in a 6-in. diameter pipe is throttled from 200 psia and 50 F to 10 psia. (*a*) What is the temperature after throttling? (*b*) Find the change in entropy per pound of air. (*c*) What is the loss of available energy? (*d*) What is the change in enthalpy per pound of air?

26. A heat engine is operated between the temperature limits of 2500 and 500 F. The engine is supplied with 10,000 Btu per bhp-hr. (*a*) What is the actual thermal efficiency? (*b*) Find the Carnot cycle efficiency.

(*a*) 25.47 per cent, (*b*) 67.6 per cent, *Ans.*

27. A quantity of air at 60 F has a volume of 0.25 cu ft and a pressure of 15 psia. This air is heated at constant volume until the pressure is 150 psia, expanded isentropically to a pressure of 15 psia, and cooled at constant pressure to the original volume. (*a*) Draw a *PV* diagram. (*b*) How much heat was added? (*c*) Find the net work done per cycle. (*d*) What is the efficiency of the cycle?

28. Assume 1 cu ft of pure air at 100 F and 14.0 psia in an engine cylinder. The air is compressed isentropically to 0.125 cu ft, heated to 2500 F at constant volume, expanded isentropically in the cylinder to its original volume, and cooled to its original pressure. (*a*) Sketch the cycle. (*b*) What is the efficiency of the cycle? (*c*) How much work is done per cycle? Assume c_p and c_v constant at 0.24 and 0.171, respectively.

CHAPTER

4

Steam and Steam Calorimetry

49. Steam Generation. When heat energy is transferred to water, its enthalpy and its physical state change. As heating takes place the temperature of the fluid rises and generally its density decreases. The rapidity of the vaporization is dependent upon the rate of heat transfer to the water and its movement in the confining vessel. The vapor thus formed is known as steam, and though it exists in a gaseous state it does not entirely follow the laws of a perfect gas.

The temperature at which boiling or vaporization occurs is dependent upon the purity of the water and the absolute pressure exerted upon it. For pure water the boiling temperature is definitely fixed for each pressure exerted upon it, and it is less at low pressures than at high. Numerical examples of pressures and the corresponding vaporization temperatures of pure water are 0.491 psia, 79.03 F; 14.696 psia, 212 F; and 600 psia, 486.2 F.

50. Saturated Steam. Steam that exists at the vaporization temperature corresponding to its (absolute) pressure is defined as *saturated.* Saturated steam may be completely free from unvaporized water particles, or it may carry water globules in suspension. Thus saturated steam may be in either a wet or a dry condition.

The generation of steam is a steady-flow process to which equation 22 is applicable. The term enthalpy is defined in Art. 10. In the production of saturated steam the absorption of energy (heat), by each pound of water, takes place in two stages which are: (1) addition of the *enthalpy of the liquid*, h_f, and (2) addition of *enthalpy of evaporation*, h_{fg}.

All materials having a temperature above −459.7 F possess energy. For water and steam a base temperature of 32 F is more convenient to use than zero absolute. Consequently, the custom is to reckon enthalpies of the liquid, evaporation, and the vapor as well as all changes of entropy per pound above the 32 F datum temperature. "Thermodynamic Properties of Dry Saturated Steam," as compiled by Professors Joseph H. Keenan and Frederick G. Keyes, are given by Tables XIV and XV in the Appendix.

51. Enthalpy of Liquid. The amount of heat energy, in Btu, necessary to raise 1 lb of water from a temperature of 32 F to its temperature of boiling, at a given absolute pressure, is the ***enthalpy of the liquid***, h_f. This quantity may be computed as the product of the mean specific heat of the water and the elevation of its vaporization temperature above 32 F. The most convenient and accurate method of determining h_f, the enthalpy of the liquid, is to make use of the data in Tables XIV and XV in the column "Enthalpy of Saturated Liquid." The enthalpies as given are the total enthalpy of the liquid reckoned above a base temperature of 32 F and are British thermal units per pound.

The amounts of heat energy added to water, as enthalpy of the liquid, increase as its pressure and temperature rise. The rate of the increase in the amount of heat added is not uniform for equal increments of pressure rise. Change of enthalpy of the liquid is used to increase the temperature of the water and to effect, at the expense of energy, small volume changes in the liquid. Consequently, when heat is added to water the major portion of it goes to increase the internal energy of the liquid, and only a very small portion is utilized in doing work. The prior discussion deals with water at its saturation temperature corresponding to the absolute pressure exerted upon it. When water is at a temperature other than that normal for the pressure its thermal properties of enthalpy of the liquid and entropy as well as its specific volume are affected by compression under a high pressure. The effects on water of such a procedure are discussed later in Art. 66, Compressed Liquid.

52. Enthalpy of Evaporation. The heat energy, in Btu, necessary to convert 1 pound of liquid water into dry steam at the same temperature and pressure is the *enthalpy of evaporation* h_{fg}. Numerical quantities for the enthalpy of evaporation are given under the heading "Enthalpy" by Tables XIV and XV. As the absolute pressure of the steam increases, the enthalpy of evaporation decreases, becoming zero at the critical pressure of 3206.2 psia, corresponding to a saturation temperature of 705.4 F. At the critical pressure water passes directly into steam without the addition of enthalpy of evaporation, and the total enthalpy of the dry saturated vapor is equal to the enthalpy of the liquid, 902.7 Btu per lb.

Enthalpy of evaporation includes two parts: the internal energy and the work, $P(v_g - v_f)/J$, done in effecting the change of the volume of 1 lb of water to the volume of 1 lb of dry steam formed at the existing absolute pressure. Both items are usually given in Btu per pound.

The work equivalent, in Btu, is

$$P(v_g - v_f)/J = 144p(v_g - v_f)/778 \tag{130}$$

where J = 778 ft-lb per Btu.
P = steam pressure, psfa.
p = steam pressure, psia.
v_g = volume of the steam formed, cu ft per lb.
v_f = volume of the water at the vaporization temperature, cu ft per lb.

The internal energy of evaporation u_{fg} is the energy, in Btu, necessary to overcome the molecular attraction of the water molecules at the vaporization temperature. This internal energy of evaporation is equal to

$$u_{fg} = h_{fg} - P(v_g - v_f)/J \tag{131}$$

Data taken from either Table XIV or XV may be used to compute u_{fg} as $u_g - u_f$ when the total internal energy of dry saturated vapor is u_g and the internal energy of its liquid is u_f.

53. Total Enthalpy of Dry Saturated Steam. The *total enthalpy* h_g of 1 lb of dry saturated steam, reckoned above 32 F, is the sum of the enthalpy of the liquid and the enthalpy of evaporation.

$$h_g = h_f + h_{fg} \tag{132}$$

where h_g = the enthalpy of the vapor, Btu per lb.
h_f = the enthalpy of the liquid, Btu per lb.
h_{fg} = the enthalpy of evaporation, Btu per lb.

The total enthalpy of dry saturated steam is dependent upon its pressure, as shown by the numerical quantities given under the heading "Enthalpy of Saturated Vapor, h_g," in Tables XIV and XV

When equation 132 is utilized, the values of h_f and h_{fg} are taken from either Table XIV or XV, depending upon the known condition; that is, temperature or pressure. Tables XIV and XV are not complete for all temperature and pressure conditions of dry saturated steam. When the properties are not given in the tables for the conditions desired, numerical data of sufficient accuracy may be obtained by interpolating between the data given for the conditions nearest to those desired.

Example. Find the total enthalpy of dry saturated steam at 175 psia.

Solution. In this case an interpolation will be necessary between the values given for 170 and 180 psia. Hence, $h_g = 1196 + [5(1196.9 - 1196) \div 10] = 1196.5$ Btu per lb. Using equation 132 and interpolating the component parts, $h_g = 343.6 + 852.9 = 1196.5$ Btu per lb.

54. Specific Volume and Density of Dry Saturated Steam. The specific volumes v_g, in cubic feet, occupied by 1 lb of dry saturated steam are given under the heading "Specific Volume" in Tables XIV and XV. The volume of 1 lb of steam varies inversely as the pressure.

The density d_g of dry saturated steam is the weight of the steam in pounds per cubic foot and is the reciprocal of the specific volume v_g. The density varies as the pressure; thus the higher the pressure, the greater is the weight of steam per cubic foot. The volumes v_f of the liquid water are also given in the steam tables, and the density of the liquid may be computed as the reciprocal of v_f.

55. Quality of Steam. Moisture present in saturated steam may be due to several causes. Steam conveyed by piping may suffer a loss of heat by convection and radiation so that part of the enthalpy of evaporation is given up and water particles are formed in the steam. The generation of the steam may have been imperfect either on account of poor design of the steam generator, high rate of evaporation maintained, or poor boiler feedwater.

The dry steam per pound of wet steam is the quality of the steam. The quality x of steam may be expressed either as a percentage or as a decimal. As most calculations involve the use of steam quality expressed as a decimal, it will always be so used in this work.

56. Total Enthalpy of Wet Saturated Steam. The total enthalpy of a pound of wet saturated steam is less than that of a pound of dry saturated steam under the same pressure conditions. When the water is not completely vaporized, the heat carried as enthalpy of evaporation is less than when the steam is dry. The quality factor affects only the enthalpy of evaporation and does not change the enthalpy of the liquid at any given pressure. The total enthalpy of a pound of wet steam reckoned above 32 F, is

$$h_{gw} = h_f + xh_{fg} \tag{133}$$

where h_{gw} = total enthalpy of wet steam, Btu per lb.
x = quality of the steam, expressed decimally.
h_f = enthalpy of the liquid, Btu per lb.
h_{fg} = enthalpy of evaporation of dry saturated vapor, Btu per lb.

57. Specific Volume and Density of Wet Saturated Steam. When the quality of steam is expressed decimally, the specific volume v_{gw} of wet saturated steam may be computed as

$$v_{gw} = v_f + x(v_g - v_f) \tag{134}$$

In the foregoing equation, v_f and v_g are obtained from the steam tables giving the properties of dry saturated steam for the pressure involved.

When available v_{fg} may replace $(v_g - v_f)$. The steam density is the reciprocal of v_{gw}.

58. Superheated Steam. The temperature and the total enthalpy of saturated steam, at any pressure, may be increased by the application of additional heat. When the steam temperature is above the saturation temperature, corresponding to the steam pressure, the steam is superheated. The superheating process usually takes place in a separate piece of equipment where the steam is not in contact with its liquid. The total enthalpy of a pound of superheated steam, designated as h, may be ascertained most readily from Table XVI. The total enthalpy of a pound of superheated steam may be computed by adding to the total enthalpy of 1 lb of dry saturated steam, at the given pressure, the enthalpy added during the superheating process. This latter quantity may be expressed as $c_{ps}(t_s - t)$, when t_s is the temperature of the superheated steam and t is the temperature of saturated steam at the existing absolute steam pressure. The mean specific heat of superheated steam c_{ps} varies with the pressure and the steam temperature. The range of values of the mean specific heat of superheated steam may be between 0.44 for small amounts of superheating at subatmospheric pressures and more than 2 for high temperatures and pressures.

The advantages of superheating steam are (1) the additional heat imparted to the vapor causes it to behave more nearly as a perfect gas than saturated steam, (2) the temperature ranges in certain pieces of steam-using equipment may be increased, and (3) steam condensation losses may be eliminated or lessened in pipe lines and steam-using engines and turbines.

59. Examples of the Use of Steam Tables. The following examples serve to illustrate calculations dealing with the properties of steam.

Example. One pound of dry saturated steam exists at a pressure of 190 psia. Find the temperature of the steam, the total enthalpy of the vapor, the specific volume and the density of the steam, the work done, and the internal energy added at the temperature of vaporization.

Solution. From Table XV the saturation temperature is 377.51 F, and the total enthalpy is 1197.6 Btu per lb. The specific volume of the vapor v_g is 2.404 cu ft per lb, and the density d_g is $1 \div 2.404 = 0.416$ lb per cu ft. The work done at vaporization is $144 \times 190(2.404 - 0\ 01833) \div 778 = 83.8$ Btu per lb; hence, the internal energy added at vaporization is the enthalpy of evaporation, 846.8, minus 83.8, which is 763 Btu per lb.

Example. Steam at 150 psia has a quality of 0.97. Find the total enthalpy of the vapor and the specific volume and the density of the steam.

Solution. The total enthalpy of 1 lb of the vapor is, using equation 133, $h_g = 330.51 + (0.97 \times 863.6) = 1168.2$ Btu per lb. As not all the water is vaporized, the specific volume of the wet steam will be $v_f + x(v_g - v_f) = 0.01809 + 0.97(3.015 -$

0.01809) = 2.925 cu ft per lb. The density of the steam is 1 ÷ 2.925 = 0.342 lb per cu ft. The volume change effected at vaporization is 0.97(3.015 − 0.01809) = 2.907 cu ft. The work done at vaporization, expressed in Btu per lb, is (144 × 150 × 2.907) ÷ 778 = 80.7 Btu per lb, and the change of internal energy at vaporization is (0.97 × 863.6) − 80.7 = 757 Btu per lb.

Example. Water at 70 F is converted into superheated steam at a pressure of 200 psia and a total temperature of 500 F. The mean specific heat of the superheated steam may be taken as 0.596 Btu per lb per deg F. Find the change of enthalpy necessary to produce 1 lb of the steam.

Solution. From Table XVI the total enthalpy of the steam may be found as 1268.9 Btu per lb, or it may be computed as $h = 1198.4 + 0.596\ (500 - 381.79) =$ 1268.9 Btu per lb. The initial enthalpy of the liquid of the water is given as 38.04 Btu lb in Table XIV. The enthalpy added per pound of water to produce the superheated steam is 1268.9 − 38.04 = 1230.86 Btu per lb.

60. Steam-Quality Determination. The dryness or quality of saturated steam is determined by means of some type of steam calorimeter. There are two general types of steam calorimeters: the *separating* and the *throttling or superheating* instruments. Combinations of these types may be used where the steam is of low quality and where a greater degree of accuracy is necessary. In all determinations of steam quality the calorimeter must be insulated against the loss of heat from the incoming steam, and also the sample of steam drawn must be representative of the steam for which the quality determination is desired.

Separating calorimeters employ mechanical removal of moisture from wet steam entering them. Measurements are necessary of the weight of moisture collected, as water, and the weight of dry steam escaping from the calorimeter. Separating calorimeters may be used for the determination of the quality of low-pressure or very wet steam. The expression of quality is, decimally,

$$x = \frac{m_{ds}}{m_{ds} + m_m} \tag{135}$$

where x = quality of steam.

m_{ds} = weight of dry steam passing through the calorimeter, lb.

m_m = weight of moisture separated from m_{ds} lb of dry steam, lb.

Throttling calorimeters operate upon the principle that, if steam is throttled and then expanded without doing any work or without the loss of heat, the total energy of the steam is unchanged. Radiation losses should be minimized by having the calorimeter, as well as its connection to the steam main, thoroughly insulated. The calorimeter shown by Fig. 35 involves the principles of the throttling calorimeter. Steam enters through the sampling nozzle and is expanded in an orifice.

The expanded steam enters the steam chamber and escapes at its top after traversing a U-path. Momentary excess moisture is separated and re-evaporated within the steam chamber, and the effect is indicated in direct proportion by the superheat thermometer placed in the path of the steam. A throttling, separating, and re-evaporating calorimeter is formed in a single chamber which is jacketed by the escaping steam. The jacket steam is protected against chilling by heavy insulation and a nickel-plated casing. A steam pressure gage is necessary to indicate the initial pressure of the steam. The exhaust takes place freely to the atmosphere.

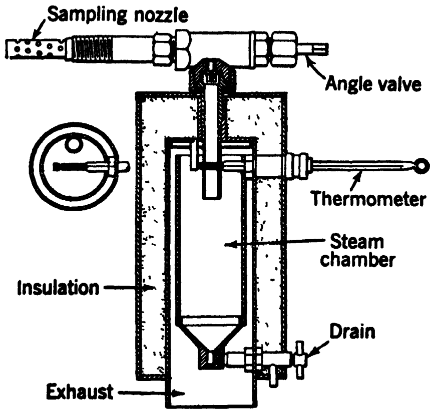


FIG. 35. Ellison steam calorimeter.

If the total energy of the steam is unchanged in the expansion process and the moisture of the steam is not excessive it will be evaporated, and the steam in the low-pressure side of the calorimeter will be superheated at its existing pressure. This relation may then be established: the total enthalpy of the wet saturated steam before expansion ($h_f + xh_{fg}$) is equal to the total enthalpy h of the superheated steam after expansion. In equation form this may be stated $h_f + xh_{fg} = h$, and

$$x = \frac{h - h_f}{h_{fg}} \tag{136}$$

where x = initial quality of the steam expressed decimally.

h_f = enthalpy of the liquid at the initial pressure abs, Btu per lb.

h_{fg} = enthalpy of evaporation of dry saturated vapor at the initial pressure abs, Btu per lb.

h = total enthalpy of the superheated steam at the calorimeter pressure absolute, Btu per lb.

For accurate work, corrections should be properly applied to the readings of thermometers having mercury columns exposed to the surrounding air. At least 10 F of superheat must be obtained by the expansion of the steam.

61. Change of Entropy. The ratio of the amount of heat energy added to or taken from 1 lb of steam to the absolute temperature at which the addition or the abstraction is made expresses the change of

entropy. The enthalpy of evaporation added isothermally to 1 lb of water at its boiling temperature, T deg F abs, produces the change in entropy at vaporization $h_{fg} \div T$ or s_{fg}. Thus the entropy change is the change in heat per degree of absolute temperature and is generally referred to as entropy. The total entropy of saturated steam is the sum of the entropy of the liquid s_f and the entropy of evaporation s_{fg}. Numerical examples for the entropy of 1 lb of dry saturated steam may be found by reference to Tables XIV and XV.

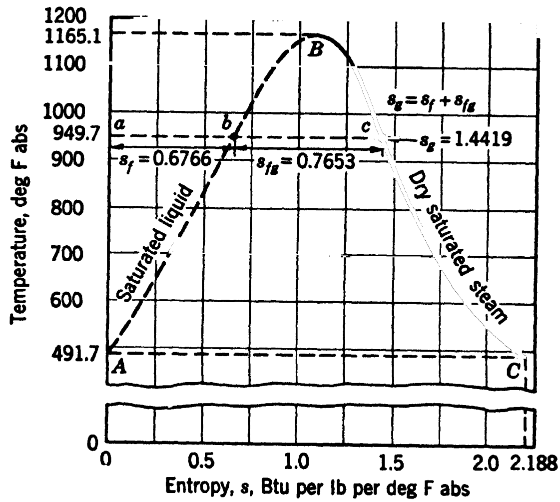


FIG. 36. Temperature-entropy diagram for dry saturated steam.

The change of entropy of the liquid cannot be expressed so simply as the entropy of evaporation because the liquid temperature is not constant and because the specific heat of water varies with different temperatures. If heat is considered as being added in infinitely small amounts dh_f at the changing absolute temperature T between the initial and final absolute temperatures T_1 and T_2, then, for any given temperature T, $ds_f = dh_f/T$, and the increment of heat is equal to $T\,ds_f$ or $c_{pw}\,dT$. The summation of the infinitely small increments for a given temperature rise or change of entropy of the liquid is

$$s_f = \int_{T_1}^{T_2} c_{pw} \frac{dT}{T} \tag{137}$$

When the specific heat $c_{pw} = 1.0$, the expression integrated becomes

$$s_f = \log_e T_2 - \log_e T_1 = \log_e \frac{T_2}{T_1} \tag{138}$$

Figure 36 is a temperature-entropy diagram for 1 lb of water converted into dry saturated steam at various absolute temperatures. The path of the plotted values of the entropy changes of the liquid s_f, at various temperatures, falls along the line AB, while the total entropy of the vapor s_g falls along the dry and saturated line CB. Thus at a temperature of 491.7 F abs the total entropy change s_g, or 2.188, is represented by the length of the horizontal line AC. In this particular case the entropy of the liquid s_f is zero, and the entropy of evaporation s_{fg} is equal to AC. For water evaporated at a pressure of 621.4 psia and temperature of 949.7 R the entropy of the liquid, a to b, is 0.6766; from b to c the entropy of evaporation is 0.7653; and the total entropy of the vapor, at c, is 1.4419. At an absolute temperature of 1165.1 F, corresponding to the critical pressure of 3206.2 psia the entropy of the liquid, 1.0580, is equal to the total entropy, as the entropy of evaporation has become zero at point B.

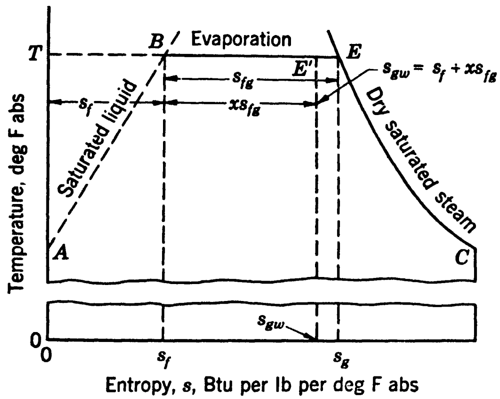


FIG. 37. Temperature-entropy diagram, wet steam.

62. Entropy Changes of Wet Steam. When steam carries moisture in suspension, the total entropy changes and those indicated at evaporation are less than those indicated in the steam tables. In the calculation of the total enthalpy of 1 lb of wet steam the quality factor affects only the enthalpy of evaporation, Fig. 37. Hence, in the calculation of the change of entropy of 1 lb of water converted into wet steam, the entropy of evaporation is multiplied by the steam quality expressed decimally. The total entropy change is

$$s_{gw} = s_f + xs_{fg} \tag{139}$$

63. Entropy Changes of Superheated Steam. When saturated steam is converted into superheated steam at the same absolute pressure

there is a further change of entropy. Additional changes of entropy s_g to s, due to superheating of steam, are shown by the curve between points E and F of Fig. 38. Entropy changes during superheating occur with increasing temperatures and varying specific heats. Consequently, the method of calculating the changes that occur during the superheating process is similar to that employed for entropy variations of the liquid. The reader is referred to Table XVI for entropy changes of superheated steam.

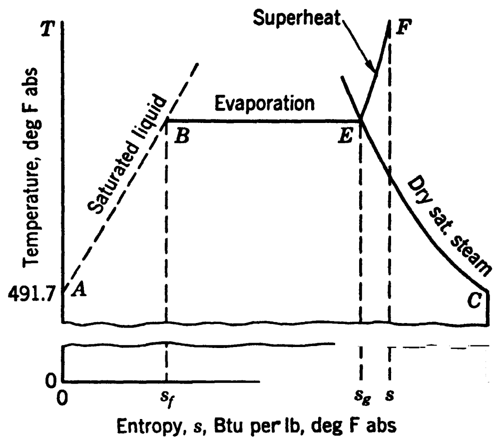


FIG. 38. Temperature-entropy diagram for superheated steam.

64. Constant Entropy. When adiabatic conditions are maintained the heat of a material is neither given to nor received from external bodies. Under reversible adiabatic or isentropic conditions there is no change in the total entropy. However, if steam expands isentropically from a condition of higher temperature to one of lower temperature and if work is done, there is a reduction of the total enthalpy of the steam. If the expansion of superheated steam is carried far enough or if the steam is initially saturated, the quality of the steam after expansion will be less than unity when work is done. If the initial conditions and the final steam pressure after isentropic expansion are known, the final quality may be computed. With no change of entropy during adiabatic expansion in which work is done, the initial entropy is equal to the final entropy.

$$s_{f,1} + x_1 s_{fg,1} = s_{f,2} + x_2 s_{fg,2}$$

$$x_2 = \frac{s_{f,1} + x_1 s_{fg,1} - s_{f,2}}{s_{fg,2}} \qquad (140)$$

The numerical quantities are taken directly from the steam tables.

Example. *Find the quality and the total enthalpy of 1 lb of saturated steam after it has been isentropically expanded to a pressure of 30 psia from an initial condition of 100 psia and 0.98 quality.*

Solution. Using data from Table XV the following are obtained:

$$x_2 = \frac{0.4740 + (0.98 \times 1.1286) - 0.3680}{1.3313} = 0.91$$

$$h_{gw,2} = 218.82 + (0.91 \times 945.3) = 1079 \text{ Btu per lb}$$

65. Total Enthalpy-Entropy Diagrams. The Mollier diagram, Fig. 39, affords a ready and rapid aid in solving problems involving the properties of steam. From this chart the total enthalpy and the entropy changes of steam at various pressures may be read for either a dry, wet, or superheated condition. The diagram permits a quick determination of the quality of steam from data given by a throttling calorimeter test. The total enthalpy and moisture content of steam may be determined after isentropic expansion, also from the chart, when the initial steam conditions are known.

Example. Find the total enthalpy and the total entropy of 1 lb of steam at a pressure of 200 psia and 0.96 quality.

Solution. Locate on the diagram, Fig. 39, the intersection of the 0.96 quality line and the 200 psia pressure line. Read along a horizontal line the total enthalpy of the steam as 1164.5 Btu per lb and along a vertical line the total entropy as 1.505.

Example. Steam at a pressure of 400 psia had a total temperature of 700 F. Find the total enthalpy and the total entropy per pound.

Solution. Find the intersection of the 400 psia pressure line and the 700 F temperature line. The total enthalpy corresponding to this intersection is 1362.8 Btu per lb, and the total entropy is 1.64.

Example. A throttling calorimeter test gave the following data: average initial steam pressure, 150 psia; calorimeter exhaust pressure, 14.7 psia; and temperature of steam in the calorimeter, 260 F. Find the quality of the steam.

Solution. Locate the intersection of the 260 F line and the 14.7 psia pressure line. Move horizontally from this point to the 150 psia pressure line, and there read a quality of 0.976.

Example. Steam at 250 psia and 0.99 quality is expanded isentropically to a pressure of 1.5 psia. Find the total enthalpy and the quality of the steam after expansion.

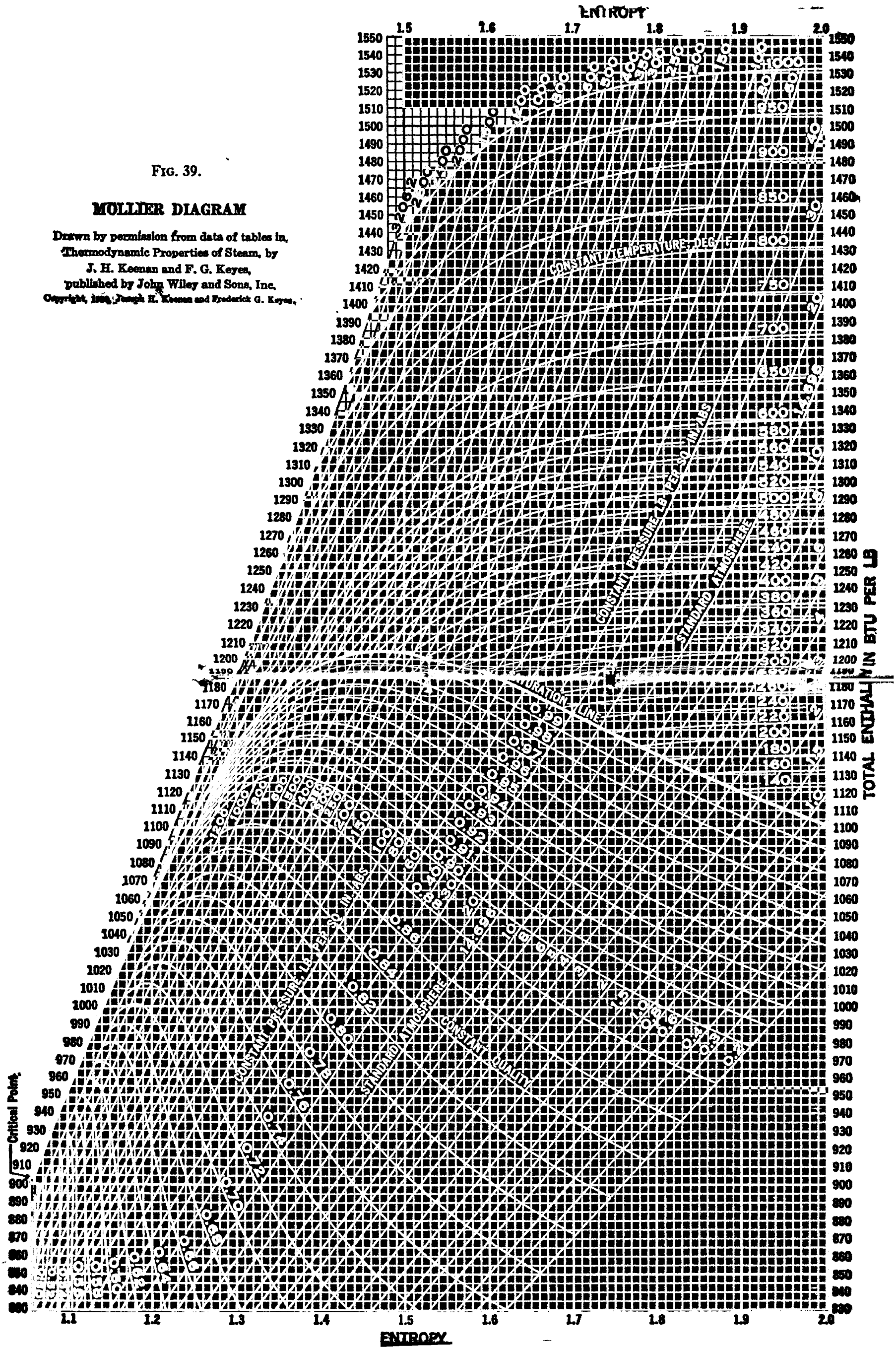
Solution. From the intersection of the 250 psia pressure line and the 0.99 quality line drop vertically along a line of constant entropy to the 1.5 psia pressure line. The quality is 0.758, and the total enthalpy is 863 Btu per lb.

66. Properties of Compressed Liquid. In the operation of a steam generator water is supplied to it under a pressure greater than that of

FIG. 39.

MOLLIER DIAGRAM

Drawn by permission from data of tables in, Thermodynamic Properties of Steam, by J. H. Keenan and F. G. Keyes, published by John Wiley and Sons, Inc.



its condition of operation, with an enthalpy less than that of the water within the unit. For pressures of 400 psia or less the data of prior-mentioned steam tables may be used with little error. For conditions of operation of high-pressure boilers attention should be given to the properties of a compressed liquid when its temperatures do not correspond to the pressures exerted upon it for conditions of saturation. The data of Table XVII of the Appendix include those for liquid water under saturation conditions at various temperatures. The corrections to be applied to secure the correct numerical quantities for volume, enthalpy, and entropy of compressed water at temperatures which are not those of saturation for various pressures exerted upon it are also given.

During the feedwater pumping operation, compression of the water occurs and its volume is slightly decreased below that of its initial condition. As the process takes place the internal energy of 1 lb of the liquid decreases, because of the lessened distance between the molecules; the flow energy increases as the increase of pressure takes place even though there is a slight volume decrease; the temperature rises slightly; the enthalpy increases; and the entropy of the liquid decreases. Interpolations between the data given in Table XVII are necessary when conditions are other than those listed.

Example. A feedwater heater delivers water at a saturation temperature of 250 F, corresponding to a pressure of 29.825 psia. The water is forced into a boiler against a pressure of 1500 psia. Find the volume of the water, its enthalpy of the liquid, and the entropy per pound under the higher pressure.

Solution. Neglect the temperature rise of the water. For a pressure of 1500 psia and a temperature of 250 F it is necessary to interpolate the correction factors between those given for temperatures of 200 and 300 F.

For conditions of saturation at 250 F and 29.825 psia, $v_f = 0.01700$ cu ft, $h_f = 218.48$ Btu, and $s_f = 0.3675$. By the processes of interpolation $(v - v_f) = -9.25/10^5 = -0.000093$, $(h - h_f) = 3.03$, and $(s - s_f) = -2.16/10^3 = -0.0022$. The corrected volume is $0.01700 - 0.00009 = 0.01691$ cu ft per lb. The corrected enthalpy is $218.48 + 3.03 = 221.51$ Btu per lb. The corrected entropy becomes $0.3675 - 0.0022 = 0.3653$ Btu per (lb)(deg F).

PROBLEMS

1. (*a*) What absolute pressure exists within a closed container which confines water and either wet or dry saturated steam maintained at a temperature of 350 F? (*b*) How much greater would the temperature of the steam in the container be if its pressure were 500 psia?

2. Does the temperature of saturated steam increase uniformly with equal increases of pressure?

3. Calculate the enthalpy of the liquid for 1 lb of steam existing at a pressure of 227 psia when the mean specific heat of water is 1.019.

4. (*a*) Why does the enthalpy of evaporation, h_{fg}, for dry saturated steam decrease as the absolute pressure of the steam is increased? (*b*) Calculate the internal energy of evaporation, u_{fg}, for 1 lb of dry saturated steam which exists at a pressure of 350 psia.

5. Calculate the heat equivalents of the internal-energy change and the work done in effecting evaporation alone of 50 lb of water existing at a temperature corresponding to a pressure of 500 psia. 33,550 Btu; 4200 Btu, *Ans.*

6. From the data of Table XV determine all the numerical values, except those for entropy, given as properties of dry saturated steam at 150.5 psig when the barometric pressure is 29.6 in. of mercury.

7. A steam boiler discharges 350,000 lb of dry steam during a period of 4 hr with these conditions of operation: steam pressure, 215 psig; barometric pressure, 28.8 in. of mercury; and feedwater temperature, 120 F. Find (*a*) the enthalpy of the liquid of the water supplied per hour at its initial temperature. (*b*) Calculate the total enthalpy of the dry steam produced per hour. (*c*) Find the energy added to the water per hour of steam generation.

8. A steam boiler produces dry saturated steam at a pressure of 275 psia. The change of enthalpy of the feedwater is 1028.5 Btu per lb. What is the temperature of the feedwater as it enters the unit? 205.45 F, *Ans.*

9. From the data of Tables XIV and XV calculate the densities of the liquid and the densities of dry saturated steam for (*a*) temperatures of 90 and 300 F, and (*b*) pressures of 275 and 650 psia.

10. Wet steam at a pressure of 240 psia has a total enthalpy of 1185.9 Btu per lb. What is the quality of the steam?

11. A steam boiler receives feedwater at a temperature of 205 F and operates to produce 100,000 lb per hr of wet steam at a pressure of 350 psia and a quality of 0.975. Find the enthalpy change produced. 101,100,000 Btu, *Ans.*

12. Steam having an initial quality of 0.99 and a pressure of 900 psia is reduced to a final pressure of 450 psia with a loss of heat amounting to 11 Btu per lb. Find the final quality of the steam and the increase of moisture per pound of steam.

13. Dry saturated steam has a total enthalpy of 1203.9 Btu per lb at pressures of 350 and 550 psia. (*a*) What loss of heat is necessary to reduce the steam to a dry saturated condition at a pressure of 250 psia. (*b*) In passing from 550 psia to dry saturated steam at 250 psia, at what pressures will the steam be wet and what is its average quality?

14. From data of Tables XIV and XV calculate the densities of the liquid and the densities of steam of 0.97 quality when (*a*) the temperatures are 400 and 560 F, and (*b*) the pressures are 160 and 1000 psia.

15. The volume of a steam space is 50 cu ft, and the space is filled with dry saturated steam at 190 psia. What is the weight of the steam in the space?

16. If the steam held in the space were of 0.98 quality, in problem 15, what weight would be involved?

17. A steam generator including a boiler and a superheater operates with feedwater at an initial temperature of 210 F, steam leaving the boiler at 825 psia and 0.99 quality, and steam leaving the superheater at a pressure of 800 psia and a temperature of 700 F. (*a*) Find the degrees of superheat held by the leaving steam. (*b*) Calculate the change of enthalpy, in Btu per pound, in the boiler, in the superheater, and in both units combined. (*c*) What change in volume, in cubic feet per pound, takes place in the superheater?

(*a*) 181.8 F; (*b*) 1012.9, 147.6, and 1160.5 Btu per lb; (*c*) 0.2379 cu ft per lb, *Ans.*

18. Calculate the mean specific heat of superheated steam between the saturation temperature and each total temperature listed in Table XVI for the pressure condition of 200 psia. Plot a curve using mean specific heats as ordinates and the total steam temperatures as abscissas.

19. The mean specific heat of superheated steam between the saturation temperature and a total temperature of 300 F for a vapor pressure of 14.696 psia is 0.482. Calculate the total enthalpy of 1 lb of the vapor, and compare the quantity with that given in the table for the stated conditions.

20. A steam boiler, equipped with a superheater, receives feedwater at a temperature of 230 F and discharges 500,000 lb per hr of steam at 800 F and 375 psia. Find (*a*) the degrees of superheat; (*b*) the mean specific heat of the superheated steam; and (*c*) the total change of enthalpy in the process.

(*a*) 361.8 F; (*b*) 0.588; (*c*) 609,650,000 Btu per hr, *Ans.*

21. For the conditions of problem 20 compute the work done, both as foot-pounds per hour and Btu per hour, in the process of converting the water into superheated steam.

22. A power plant has steam available at a pressure of 750 psia and a total temperature of 660 F. The plant has a steam turbine which is capable of using steam at a pressure of 400 psia. Under what temperature or quality condition will steam reach the turbine if the 750 psia steam is reduced to 400 psia without loss of heat?

23. The flow and pressure gage of a separating calorimeter indicates a discharge of 2.8 lb of dry steam per 10 min. The moisture collected within the calorimeter during the 10-min. interval is 0.15 lb. Compute the quality of the steam.

24. A separating calorimeter is operated without a dry-steam flow gage. The escaping dry steam is reduced to water in a small condenser, and a quantity of 3.0 lb is weighed during a period of 20 min. The moisture separated from the steam during the test is 0.30 lb. Find the quality of the steam.

25. A throttling calorimeter receives steam at a pressure of 260 psia and discharges it at a temperature of 300 F against a pressure of 20 psia. What was the initial quality of the steam?

26. A throttling calorimeter test gives the following data: barometric pressure, 14.2 psia; steam temperature in the calorimeter thermometer chamber, 300 F; and initial steam pressure, 165.8 psig. Compute the quality of the steam.

27. Steam of 0.97 quality is supplied to and handled in a calorimeter having the same pressure operating conditions as stated by problem 26. What is the temperature of the steam in the calorimeter?

28. What is the minimum quality of steam which may be determined by use of a throttling calorimeter which functions with only 10 deg of superheat in the escaping steam with the following conditions of operation: mean specific heat of superheated steam, 0.49; initial steam pressure, 195 psia; and thermometer-chamber steam pressure, 20 psia.

29. A throttling calorimeter discharges steam at 20 psia and a total temperature of 303 F. At what initial pressure could the supply steam have been dry and saturated?

30. A throttling calorimeter is constructed of pipe fittings and an orifice plate. The test instruments available are two thermometers of suitable ranges of graduations and an aneroid barometer. With these pieces of equipment the following data are taken: initial temperature of the wet steam, 350 F; temperature of the exhaust steam, 230 F; and barometric pressure 29.92 in. of mercury. Compute the steam quality.

0.962, *Ans.*

31. Check by numerical calculation the quantity stated in the column headed

"Entropy of Saturated Liquid" of Table XIV for a steam temperature of 100 F.

32. Compute the change of entropy for the liquid of saturated steam through the temperature range of 32 to 200 F.

33. By use of Table XV ascertain the entropy change at evaporation for dry saturated steam at the following pressures: 65, 150, 400, and 950 psia. Check the foregoing quantities by numerical calculations.

34. By the use of Table XIV secure data for the entropy changes at evaporation of saturated water vapor for the following temperatures: 200, 400, 600, and 700 F. Compute the required data.

35. Steam is isentropically expanded from 225 psia and 450 F to a pressure of 16 psia. Compute the final quality and the enthalpy of the steam.

36. Find the quality and the total enthalpy of steam at a terminal pressure of 18 psia after it has been isentropically expanded from the initial conditions of 170 psia and 0.97 quality.

37. Saturated steam at a temperature of 460 F and a quality of 0.96 is isentropically expanded to a final temperature of 160 F. Find the final steam quality and the total enthalpy per pound of steam.

38. Find, by use of a Mollier diagram, the total enthalpy for steam at the following conditions: (*a*) 250 psia and 500 F, and (*b*) 250 psia and 0.95 quality.

39. Steam is at an initial pressure of 200 psia and 480 F. Find by use of a Mollier diagram the quality and the total enthalpy of the steam after it has been expanded isentropically to a terminal pressure of 1 psia.

40. Steam at 350 psia and 460 F is expanded isentropically to a pressure of 0.8 psia. Find, by use of a Mollier diagram, the total enthalpy before and after expansion and the final steam quality.

41. Calculate the specific volume, the enthalpy, and the entropy of the liquid for 1 lb of water at a pressure of 2500 psia and a temperature of 425 F.

CHAPTER

5

Fuels and Combustion

67. Forms of Fuels. Commercial fuels, either in their natural or prepared forms, exist as solids, liquids, and gases. Solid fuels include coals, lignites, cokes, woods, and combustible wastes from many processing operations. Liquid fuels include alcohol, petroleum oils, their distillates, and sometimes tars. Natural gases come from the earth, and manufactured gases are products obtained mainly from coal. The principal combustible elements in any fuel are carbon (C) and hydrogen (H). Sulfur (S) is a combustible element but is not sought as a fuel and is considered undesirable when found in a fuel.

68. Coal. Coal is a mixture of carbon, hydrogen, oxygen, nitrogen, sulfur, moisture, and ash which once existed primarily as vegetable matter. This material became buried in marsh and lake beds. Owing to the effect of time, pressure, and temperature the vegetation has been transformed into coal. All stages of development are to be found, from the shiny structure of anthracite to the soft, crumbly forms of lignite. Many forms of vegetation can be recognized in the structure of coal. A lump of coal on display by the Illinois State Geological Survey, Urbana, Illinois, consists of a large tree stump in which the root system and grain structure are well preserved. Coal is one of the major natural resources of the world. As a fuel and as a source of chemical products its value cannot be estimated. Figure 40 shows the major coal fields of continental United States.

69. Coal Sizes. Coal as taken from the mine ranges in size from fine dust to large lumps. Generally the fine sizes contain more impurities and dirt, owing to segregation; consequently, they do not usually have so high a heating value per pound as the larger sizes. Also, special equipment may be necessary in order to burn them. Therefore, the small sizes are lower in price and find a large use for the production of steam in power plants. The preparation and sizing of coal form an essential part of the industry of coal production. The sizing operations are carried on by crushing and screening the coal. In the latter process a designated size of coal passes through the screen opening of the larger

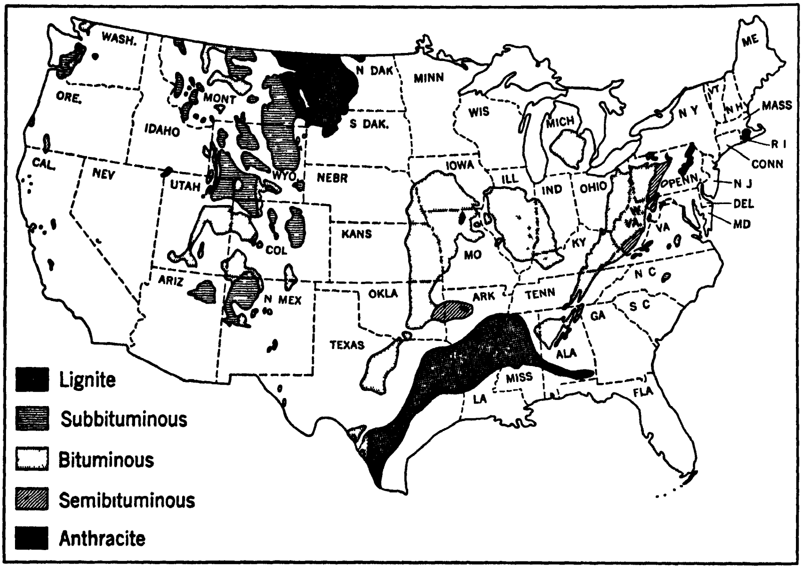


FIG. 40. Major coal fields of the United States.

specified size and over the openings of the next smaller specified size. In commercial practice considerable variation of coal sizes exists in the various coal-producing districts.

70. Coal Classifications. Coals can best be compared when placed on a combustible basis since the moisture and ash content vary widely. Figure 41 gives a graphical comparison of coals of various ranks as defined by the ASTM (American Society for Testing Materials) standards. Owing to the nature of coal no fully satisfactory classification can be made. The following subdivisions are generally used:

Anthracite is a very hard coal having a shiny black luster; it is noncoking and has a high percentage of fixed carbon and less than 8 per cent volatile matter. The coal is slow to ignite unless the furnace temperature is high, and it requires a strong draft. Anthracite coal burns either without flames or with very short bluish flames and is desirable where smokeless combustion is essential.

Semianthracite coal occurs in small quantities in various localities of the United States and is not used commercially as a steam coal. The principal differences from a true anthracite are that it has less fixed carbon, 8 to 14 per cent volatile matter, less luster, and it burns with longer and more luminous flames.

Semibituminous coal, the highest grade of bituminous coal, burns with a very small amount of smoke. This coal is softer than the anthra-

cites, contains from 14 to 22 per cent volatile matter, and has a tendency to break into small sizes during storage or transportation. The last item is of small consequence in the generation of steam so long as the size is uniform.

Bituminous coals are "soft," have high percentages of volatile matter, burn with long yellow and smoky flames, and vary greatly in the percentages of volatile matter, moisture, ash, and sulfur. The chief distinction between the eastern bituminous coals and those of the central portion of the United States is in their percentages of volatile matter

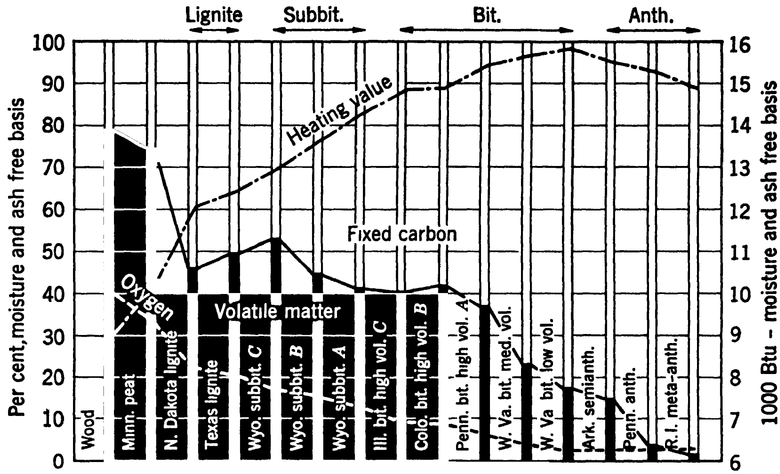


FIG. 41. Diagram showing progressive change from vegetal matter in coals of various ranks as based on ASTM classification.

and moisture when mined. Eastern coals have from 25 to 35 per cent volatile matter and from 2 to 4 per cent vein moisture. The western bituminous coals include 35 to 45 per cent volatile matter and 6 to 17 per cent vein moisture. Both high- and low-ash coals are included. A high-ash coal is usually designated as one having more than 10 per cent ash. Bituminous coals are classified as free-burning and caking or coking. Free-burning coals do not swell and fuse together when ignited, as do the coking or caking coals.

Subbituminous coals, sometimes known as black lignites, are low-grade bituminous coals which have lost the woody structural appearance of lignites. Subbituminous coals disintegrate when exposed to the air and require careful attention during storage. Their volatile matter ranges from 35 to 45 per cent and the vein moisture from 17 to 20 per cent.

Lignites are in the transition state between peat and the subbituminous grade of coal and have a woody or often a claylike appearance, together with low heat values and high moisture and high ash contents. The initial moisture may be as high as 30 to 45 per cent, but moisture is readily lost upon exposure to the air, which may produce rapid disintegration of the fuel. Unless carefully stored, lignites are subject to spontaneous combustion. Fuels of this class are of value for local use only.

Coals vary greatly in the percentages of their various constituents and consequently in the heat that they will produce per pound. This variation is marked not only in the same coal field but also in the same vein and even in the same mine.

71. Ultimate Analysis of Coal. Since coal, as found, is neither a pure substance nor of uniform composition, a definite chemical formula for it cannot be written. Consequently, a complete chemical breakdown called an ultimate analysis is necessary, if its composition is to be known. This requires the work of a chemist. The analysis and heating value of coal may be reported in any of three ways: (1) *as received or as fired*, (2) *dry or moisture free*, and (3) *ash and moisture free or on a combustible basis*. The last named assumes that everything except ash and moisture is burnable material. The chemist reports the ultimate analysis as percentages by weight and under the following headings.

MOISTURE. Moisture in coals exists as (1) surface moisture from weather and from washing operations and (2) inherent moisture, which is chemically bound within the coal structure. Surface moisture is determined by drying a sample at 90 F until the loss in weight does not exceed 0.10 per cent per hour. This preliminary drying is mainly to put the sample in equilibrium with air so that no moisture loss will occur in further testing. Inherent moisture is determined by using a finely ground sample of air-dried coal and noting the loss in weight when it is placed in an oven at 220 F for 1 hour. Total moisture is the sum of the surface and inherent moisture.

TOTAL CARBON. Total carbon is the sum of the fixed carbon and that in the volatile matter. Fixed carbon, or uncombined carbon, is that which does not distil easily when heated; it constitutes the carbon in coke. The volatile matter, C_xH_y, is liberated when coal is heated; values of x and y vary widely.

HYDROGEN. Hydrogen may exist as one constituent of the volatile matter and in the inherent moisture. The chemist's analysis will break down any unevaporated moisture into hydrogen and oxygen.

OXYGEN. All oxygen in the coal is assumed to be in combination

TABLE III

ANALYSES OF COALS OF THE UNITED STATES

(Abstracted from the U. S. Bureau of Mines, *Bulletin* 22)

Rank	State	County	Sample	Proximate Analysis, Coal as Received, Percentages: Moisture	Volatile Matter	Fixed Carbon	Ash	Ultimate Analysis, Coal as Received, Percentages: Sulfur	Carbon	Hydrogen	Oxygen	Nitrogen	Heating Value by Calorimeter (Btu per lb): As Received	Dry
Anthracite	Colorado	Gunnison	Mine	2.70	3.32	88.15	5.83	0.80	85.38	2.98	1.19	1.12	14,099	14,490
Anthracite	Pennsylvania	Lackawanna	Car culm	5.41	7.02	71.79	15.78	0.74	72.65	2.50	2.15	0.77	12,047	12,737
Anthracite	Pennsylvania	Schuylkill	Mine	2.80	1.16	88.21	7.83	0.89	84.36	1.58	1.91	0.63	13,298	13,682
Semianthracite	Colorado	Gunnison	Mine	4.86	6.96	81.87	6.31	0.81	79.66	3.43	3.00	1.93	13,468	14,155
Semianthracite	Pennsylvania	Sullivan	Mine	3.38	8.47	76.65	11.50	0.63	78.43	3.20	1.86	1.00	13,156	13,617
Semibituminous	Maryland	Allegany	Car run of mine	3.42	17.65	71.84	7.09	0.84	80.17	4.34	2.69	1.45	14,162	14,665
Semibituminous	Pennsylvania	Cambria	Car run of mine	3.51	16.82	73.04	6.63	0.94	80.70	4.17	2.79	1.26	14,279	14,800
Semibituminous	West Virginia	Fayette	Car run of mine	2.52	21.28	68.20	8.00	0.68	78.71	4.45	4.33	1.31	13,997	14,359
Semibituminous	West Virginia	McDowell	Composite mine	3.50	15.00	78.30	3.20	0.60	84.39	4.20	2.73	1.38	14,800	15,330
Bituminous	Alabama	Bibb	Mine	3.16	31.05	59.56	6.23	1.20	78.28	4.98	4.78	1.37	14,141	14,602
Bituminous	Colorado	Boulder	Car run of mine	18.68	34.88	40.45	5.99	0.55	57.46	3.99	12.18	1.15	10,143	12,472
Bituminous	Illinois	Franklin	Mine	11.50	26.70	52.67	9.13	0.60	65.40	4.01	8.28	1.08	11,410	12,893
Bituminous	Illinois	Sangamon	Car run of mine	16.00	32.41	37.82	13.77	4.05	53.89	3.77	7.63	0.91	9,940	11,833
Bituminous	Illinois	Williamson	Mine	7.88	31.20	49.89	11.03	2.99	66.50	4.25	6.30	1.04	11,741	12,746
Bituminous	Indiana	Green	Car run of mine	10.30	36.31	41.64	11.75	4.23	61.00	4.24	7.42	1.06	11,218	12,508
Bituminous	Iowa	Marion	Car run of mine	14.21	33.17	37.40	15.22	4.66	54.08	3.93	6.59	1.31	10,019	11,678
Bituminous	Kansas	Crawford	Car run of mine	4.99	32.68	49.36	12.97	4.28	67.34	4.43	4.91	1.03	12,242	12,884
Bituminous	Kentucky	Hopkins	Car run of mine	7.92	36.09	45.93	10.06	3.52	65.29	4.51	7.30	1.40	12,022	13,055
Bituminous	Missouri	Clay	Composite mine	15.83	32.80	41.46	9.91	2.97	58.62	4.05	7.54	1.08	10,622	12,619
Bituminous	Ohio	Hocking	Mine	9.72	32.44	53.41	4.43	0.54	69.50	4.62	9.94	1.25	12,247	13,567
Bituminous	Oklahoma	Pittsburg	Mine	2.09	27.59	50.25	20.07	5.73	63.66	4.23	2.89	1.33	11,695	11,945
Bituminous	Pennsylvania	Fayette	Car run of mine	5.13	27.87	58.29	8.71	0.86	73.13	4.34	6.33	1.50	13,365	14,089
Bituminous	Pennsylvania	Westmoreland	Car run of mine	3.39	31.79	56.46	8.36	1.05	74.42	4.69	6.70	1.39	13,699	14,178
Bituminous	Tennessee	Campbell	Car run of mine	5.38	34.54	53.03	7.05	0.99	72.41	4.90	7.55	1.72	13,048	13,788
Bituminous	Virginia	Lee	Car run of mine	4.06	34.93	56.28	4.73	1.20	76.59	4.87	7.31	1.24	13,826	14,411
Bituminous	West Virginia	Kanawha	Car run of mine	2.82	32.20	56.95	8.03	1.38	76.74	4.85	4.84	1.34	13,766	14,166
Bituminous	West Virginia	Mingo	Car run of mine	2.86	33.23	58.08	5.83	0.67	78.38	4.69	6.14	1.43	14,105	14,521
Sub-bituminous	Colorado	Boulder	Mine	19.15	30.82	44.27	5.76	0.25	56.38	3.80	13.58	1.08	9,616	11,893
Sub-bituminous	Washington	King	Mine	17.51	31.17	38.55	12.77	0.37	52.11	3.90	12.26	1.08	9,331	11,312
Lignite	North Dakota	McLean	Lump	35.96	31.92	24.37	7.75	1.15	41.43	2.54	9.96	1.21	7,069	11,038
Lignite	North Dakota	Williams	Car run of mine	38.92	25.54	30.15	5.39	0.48	39.34	2.56	12.63	0.68	6,739	11,034
Lignite	Texas	Wood	Car run of mine	33.71	29.25	29.76	7.28	0.53	42.52	3.04	12.13	0.79	7,348	11,084
Lignite	Wyoming	Sheridan	Mine	29.33	28.58	36.42	5.67	1.21	45.09	3.01	14.80	0.89	7,783	11,012

with hydrogen as moisture; consequently the breakdown of water vapor will account for the oxygen content.

NITROGEN. A small amount of nitrogen is present in almost all coals. Being inert, it produces no harmful effects except that its weight must be handled in the tonnage bought and it absorbs energy when passing through an engine or furnace.

SULFUR. Most coals contain some sulfur, which can be determined only by chemical analysis.

ASH. The residue left after the sample is fired at 2000 F until there is no further loss in weight is called ash.

The ultimate analysis of coal is necessary for combustion calculations. Data on many typical coals have been compiled by the U. S. Bureau of Mines and the various state geological survey institutions. Analyses of samples of the various ranks of coal, selected and adjusted from the U. S. Bureau of Mines, *Bulletin* 22, are given in Table III.

72. Proximate Analysis of Coal. A separation of coal into its physical components is called a proximate analysis and can be made without a knowledge of analytical chemistry. A proximate analysis is made by means of a chemical balance and an oven with temperature control. The components in the proximate analysis are as follows:

MOISTURE. The determination of moisture is made as for the ultimate analysis, Art. 71.

VOLATILE MATTER. The combination of carbon, hydrogen, and other gases which distil easily makes up the volatile matter. The dried sample is placed in a covered crucible and held at 1750 F for a period of 7 minutes. The loss of weight is determined after the sample is cooled, and the percentage of volatile matter is computed with account being taken of the original moisture in the coal.

ASH. The noncombustible residue is determined as for the ultimate analysis; i.e., the sample is fired at 2000 F until there is no further loss in weight. The percentage is adjusted to the original moisture content of the coal.

FIXED CARBON. The percentage of fixed carbon is determined by subtracting from 100 the sum of the percentages of moisture, volatile matter, and ash. Some of the sulfur content may vaporize with the volatile matter, and some may be included with the fixed carbon, which is obtained by a difference.

73. Coal Impurities. Moisture, ash, sulfur, oxygen, and nitrogen are coal impurities. These materials, either individually or collectively, have a marked influence on the desirability of coals for certain purposes.

Moisture has no fuel value. A loss of heat energy occurs in the

evaporation of moisture and its escape with the products of combustion. *Ash* reduces the heating value of fuel, and its removal from furnaces is expensive. It is an effective insulator against heat transfer and tends to clog the air spaces in a fuel bed, thus reducing the rate at which the fuel may be burned. Also, certain coals have ash contents which fuse at relatively low temperatures and form clinkers. Clinkers cause deterioration of furnace linings and expense in removal.

Sulfur, unless it occurs in a free state, does not have its full heating value. In the sulfate form sulfur is worthless as fuel. In large quantities it is likely to produce a troublesome clinker, and its products of combustion may produce metal corrosion. *Oxygen* has the effect of lowering the heating value of a fuel. *Nitrogen* is chemically inert, and, like ash, it decreases the energy per ton of coal.

74. Coal Storage. Users of large amounts of coal are compelled to provide facilities for storing quantities of coal for use in periods of heavy consumption or market shortages. Anthracite coal may be stored without trouble. With other coals, storage difficulties may be encountered, which are the effects of weathering upon the heating value, disintegration of the coal, and the likelihood of fire due to spontaneous combustion.

The suitability of a coal for storage purposes is influenced largely by the impurities contained within the coal and also by the size of the coal. Safety in storing coal may be achieved by the proper selection of the coal, by storage under water, or by proper piling. When stored in piles the coal should be tightly compacted and the fines mixed in with coarser material; there should not be regions of coarse material adjacent to regions of fine material; air pockets and air circulation within the pile, as well as all sources of heat, should be avoided. The piles of coal should not be made too deep, and the temperatures of their interiors should be watched closely, as excessive or dangerous temperatures will necessitate repiling. Coals high in oxygen and sulfur are harder to store satisfactorily than those having small amounts of each.

75. Pulverized Fuels. Coal, coke, and lignites when finely crushed and ground to a powder of the fineness of dust will float in the air and may be burned somewhat like a gaseous fuel. The recommended limits of fineness are that all the powdered fuel will pass through a 50-mesh screen, and 75 to 80 per cent through a 200-mesh screen. The foregoing values vary somewhat with the fuel used. In general, the finer coal is ground the better will be its combustion. However, the extra cost of very fine pulverization may exceed the value of the benefits secured so that there is an economical fineness as a limiting factor. In Art. 119 pulverized coal-burning equipment is discussed.

76. Wood, Coke, and By-Product Fuels. *Wood* as a fuel is not of general commercial importance, and its use is confined to more or less isolated conditions. The moisture content of wood is variable, being as high as 50 per cent when freshly cut and from 10 to 20 per cent after having been exposed to the air for some time. Woods are classified as *hard* and *soft*, the heat values ranging from 8300 to 9000 Btu per lb of combustible.

In the lumber industries refuse wood used for fuel includes sawdust, slabs, shavings, and "hogged" wood. The last named is wood refuse that has been cut to small chip size by a high-speed shredding machine. The hogged wood can then be blown through a conveyor pipe to the furnace. Any combustible refuse from industrial plants can be utilized as fuel; however, most waste fiber can be utilized more profitably as material for fiber board.

Coke is the solid residue left after the destructive distillation of certain kinds of coals. The combustible coke is the fixed carbon that has remained after the gases have been driven off. It is a clean, smokeless fuel which can be produced by several processes.[1]

Briquettes are made from fine coal or coke by compressing the material into small lumps with extreme pressure. A binding agent such as tar may be used. Briquettes usually command a premium price that limits their use to domestic applications.

77. Natural Gases. Methane (CH_4) and ethane (C_2H_6) are the main constituents of natural gas. They are very volatile, and, although they may be reduced to a liquid state or converted to other fuels, it is not always economical to do so. Natural gas is distributed mainly by pipe lines directly from the wells. Bulk storage is impracticable, owing to the large specific volume, except in natural underground caverns, porous rock formations, or abandoned mines. Natural gas is used largely for domestic purposes, but changes in seasonal demand make it available at times as a cheap fuel for power plants. It can be used alternately or simultaneously with oil for internal-combustion engines or with oil or powdered coal in steam generators. Gaseous fuels are ideal for furnace firing for the following reasons: (1) they contain no ash or residue; (2) they are easily mixed with oxygen, and therefore little excess air is needed; (3) they are adaptable to automatic controls; and (4) they are quickly responsive to load demands, thereby reducing stand-by losses. Table IV shows the composition of several combustible gases.

78. Liquefied-Petroleum Gases. Liquefied petroleum (LP gas) is primarily propane (C_3H_8) and butane (C_4H_{10}). These natural

[1] *Kent's Mechanical Engineers' Handbook*, 12th ed., Power Vol., John Wiley & Sons, 1950.

TABLE IV

Composition of Combustible Gases*

Constituents	Percentages, by Volume									
							Producer Gas from			
	Natural Gas	Ordinary Coal Gas	Coke-Oven Gas	Water Gas	Oil Gas	Carbureted Water Gas	Anthracite	Bituminous Coal	Coke	Blast-Furnace Gas
Hydrogen	3.0	46.0	50.0	48.0	32.0	40.0	20.0	10.0	10.0	1.0
Methane	92.0	40.0	36.6	2.0	48.0	25.0	.	3.0	.....	.....
Illuminants	3.0	5.0	4.4	. .	16.5	8.5		0.5	.	.....
Carbon monoxide	....	6.0	6.0	38.0	...	19.0	25.0	23.0	29.0	27.5
Carbon dioxide		0.5	0.5	6.0	..	3.0	5.0	5.0	4.5	11.5
Nitrogen	2.0	2.0	2.0	5.5	3.0	4.0	49.5	58.0	56.0	60.0
Oxygen	.	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	
Total	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0
Btu per cu ft†	1000	660	603	295	846	575	144	144	123	91
Volumetric air fuel ratio	9.73	6.0	5.6	2.35	8.07	5.25	1.12	1.2	0.98	0.72

* Analyses are typical, since composition of the gases is variable.
† Standard conditions, 60 F and 14.696 psia.

gases at normal temperature can be liquefied at pressures less than 200 psia. Such gases are excellent fuel for all purposes but usually are more expensive than fuel oil because of the cost of handling and storing under pressure. Liquefied-petroleum gas tanks and carburetors are available for automotive use, and some large-fleet operators have adopted LP gas for their trucks or busses. The advantages claimed are (1) easy starting in cold weather due to high volatility, (2) greater efficiency since the fuel will withstand high compression, and (3) a clean engine and lubrication system because the fuel does not condense in the crankcase. Refinery production trends indicate that LP gas will become more abundant in the next decade.

79. By-Product Gases. Some production operations yield usable gases. *Blast-furnace gas* is recovered from smelting operations in which air is forced through layers of coke and iron ore. After filtering, it may be blended with richer gas or used directly in gas engines.

Coke-oven gas results from the heating of bituminous coal to produce coke. The volatile content (8000 to 12,000 cu ft per ton) of coal is driven off by heating, and about half of this gas is used for heating the ovens. Many valuable chemical products are reclaimed as the gases are filtered and processed to remove sulfur, tar, and condensable vapors.

Sewer gas is obtained from the fermentation and decay that occur in sewage disposal vats. Consisting mainly of marsh gas (CH_4), it is

collected at large disposal plants and used as fuel for the gas engines that drive the plant pumps and agitators.

80. Producer Gases. Producer gases result from the partial oxidation of coal, coke, or peat when they are burned to destruction with an insufficient amount of air. Carbon monoxide (CO) is produced rather than carbon dioxide (CO_2); however, the carbon monoxide is diluted by the nitrogen in the air until the gas has a low-energy content, 100 to 150 Btu per cu ft. Like coke-oven gas, it must be thoroughly filtered before being used in a gas engine. It is produced in specially designed retorts. In general, producer gas is a suitable fuel only in large installations.[2]

81. Water or Illuminating Gas. Water gas is produced by blowing steam into white-hot coke or coal. The steam is decomposed, liberating free hydrogen, and the oxygen in the steam combines with the carbon to form carbon monoxide ($C + H_2O \rightarrow CO + H_2$). If coal is used, the gas composition varies as the hydrogen content. The fuel bed must be reheated frequently, and a blast of air is necessary to produce oxidation; during this process the gas is wasted.

82. Petroleum. The natural mineral oil that exists at varying depths and in many localities on the earth is called *petroleum*. It is a mixture of thousands of hydrocarbons, each having its own physical and chemical properties and molecular form. The composition ranges from very light hydrocarbons, such as CH_4, which distil rapidly at normal earth temperatures, to the heavy components, such as $C_{62}H_{126}$, which are scarcely fluid at room temperature. The boiling point differs for practically all the components. The early refineries consisted of simple evaporators, and only that fraction which had the desirable distillation temperature was condensed.

Fractional distillation is the primary method of petroleum refining; however, many special processes have been developed. *Catalytic cracking* subjects oil to high pressure and high temperature in the presence of a catalyst. The heavy oil is subdivided into lighter components. *Hydrogenation* is the process of catalytic cracking in a hydrogen atmosphere, whereby unsaturated hydrocarbons absorb hydrogen. *Polymerization* makes use of high pressure, high temperature, and a catalyst to combine light, volatile gases into gasoline. *Thermal cracking* changes heavy oils into gasoline by use of high pressure, high temperature, and exposure time.

83. Properties of Hydrocarbons. Table V shows a few of the many hydrocarbons that are known and lists some of the important properties by which they are judged as fuels.

[2] *Kent's Mechanical Engineers' Handbook*, 12th ed., Power Vol.. John Wiley & Sons, 1950.

TABLE V

PROPERTIES OF HYDROCARBONS

Family		Name	Formula	Temperature Deg F: Melting	Boiling	SIT†	Gravity: Specific	API	Heating Value* Btu: Higher per lb	Lower per lb	Mixture per cu ft	Latent Heat Btu per lb	Octane Rating	Performance Number (4ml TEL per gal): Lean‡	Rich	Mole Wt
Paraffins C_nH_{2n+2} Saturated	Gas	Methane	CH_4	−296	−258	1346	0 424	202 5	23 850	21 597	87 0	248	110‡			16 0
		Ethane	C_2H_6	−278	−127	1050	0 546	194 0	22 400	20 597	92 3	175	104‡			30 0
	LP gas	Propane	C_3H_8	−304	−44	995	0 582	142 0	21 650	20 015	93 7	167	100			44 0
		Butane	C_4H_{10}	−211	31	961	0 570	116 5	21 300	19 795	94 8	165	92			58 1
		Pentane	C_5H_{12}	−201	97	933	0 626	94 5	21 100	19 597	95 3	161	61	63	63	72 1
	Gaso line	*n* Heptane	C_7H_{16}	−131	210	893	0 684	75 5	20 850	19 408	96 4	132	0	0§	0§	100 2
		Triptane	C_7H_{16}	−13	178		0 690			19 100		125		200	360	100 2
		Iso Octane	C_8H_{18}	−162	211	1350	0 692	73 5	20 580	19 159	95 5	128	100	153	153	114 2
	Fuel oil	Decane	$C_{10}H_{22}$	−22	345	866	0 730	62 5	20 600	19 205	96 6	108				142 3
		Dodecane	$C_{12}H_{26}$	14	421		0 749	57 5	20 550	19 173	96 9	107				170 3
		Hexadecane	$C_{16}H_{34}$	65	536		0 774	51 5	20 420	19 066	96 9		100			226 4
		Octadecane	$C_{18}H_{38}$	82	586		0 782	49 5	20 400	19 047	97 3					245 5
Olefins C_nH_{2n} Unsaturated Chain		Propene	C_3H_6	−301	−54		0 61	103 0	21 000	19 450	96 5		85			42 1
		Butene 1	C_4H_8	−319	20		0 625		20 900	19 350	97 0		82	84		56 1
		Hexene 1	C_6H_{12}	−216	146		0 675	76 0	19 050	17 763	96 0	167	84 1			84 1
Naphthenes C_nH_{2n} Saturated ring		Cyclopentane	C_5H_{10}	−138	121		0 746	56 7	18 780	17 494	94 1		82	100	>160	70 1
		Cyclohexane	C_6H_{12}	44	177		0 778	51 6	18 710	17 432	94 1	156	77	84	130	84 1
Aromatics C_nH_{2n-6} Unsaturated ring		Benzene	C_6H_6	42	177	1363	0 88	29 0	18 160	17 190	96 8	169	110‡	68	>160	78 1
		Toluene	C_7H_8	−139	231	1492	0 87	31 0	18 300	17 460	99 0	156	104‡	95	>160	92 1
		Xylene	C_8H_{10}	−15	285		0 86	31 0	18 500	17 500	97 5	145	105‡			106 2
Alcohols		Methanol	CH_4O	−144	149		0 792	46 4	9 770	8 644	90 0	502	98	<75	>180	32 0
		Ethanol	C_2H_6O	−179	172		0 785	47 1	12 780	11 604	93 8	396	99	<75	>180	46 0
Tetraethyl lead			$C_8H_{20}Pb$	−213	360		1 653					73				
Hydrogen (gas)			H_2							4 300						2 0
Water			H_2O	32	212		0 998			0		970				18 0
Carbon (solid)			C							14 600						12 0
Gasoline (straight run)				−76	110–300					19 000			140‡	78	93	
Carbon monoxide			CO		−313	1128			4 347	4 347						28 0

* Constant volume at 77 F and 14 696 psia

† The self ignition temperature (SIT) is that temperature at which the fuel will ignite under compression without the aid of an open flame or spark

‡ Approximate

§ No tetraethyl lead

Most hydrocarbons, like water, can be frozen, melted, or boiled. The temperature at which these changes of phase occur are very important to aircraft, which encounter wide ranges of temperature and barometric pressure. Excessive evaporation can cause serious trouble as "vapor-lock" and loss of fuel from vented tanks.

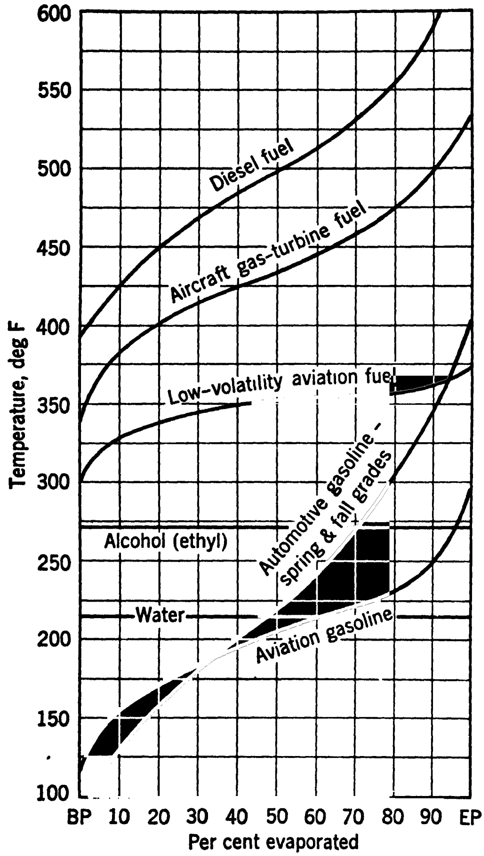


FIG. 42. Typical ASTM distillation characteristics for various types of fuels.

84. Gasolines. Gasoline is a mixture of liquid hydrocarbons which fall within specified limits. Consequently, there are "good" and "bad" gasolines. Although the Btu content varies only slightly, the difference in grade may be due to any of several properties. Table VI shows typical specifications for military and commercial aircraft gasolines.

Volatility (items 1–7, Table VI) refers to the percentage of fuel by volume that will evaporate at a given temperature. Evaporation data are shown in Fig. 42. Since gasoline is a mixture of liquid hydrocarbons,

TABLE VI

Military and Commercial Aircraft Gasoline*

No	Specification	MIL F 5572 MILITARY				ASTM D910-48T COMMERCIAL AIRCRAFT				
	Grade	80†	91/96	100/130	115/145	80/87	91/98	100/130	108/135	115/145
1	At least 10% evaporated at F	167	167	167	167	158	158	158	158	158
2	At most 40% evaporated at F	167	167	167	167					
3	At least 50% evaporated at F	221	221	221	221	221	221	221	221	221
4	At most 90% evaporated at F					212	212	212	212	212
5	At least 90% evaporated at F	275	275	275	275	257	257	257	257	257
6	Sum of 10% + 50% temp min F	307	307	307	307	307	307	307	307	307
7	End point max F	338	338	338	338	338	338	338	338	338
8	Lead ml/US gal max	0 5	4 6	4 6	4 6	0 5	2 0	3 0	3 0	4 6
9	Color	Red	Blue	Green	Purple	Red	Blue	Green	Brown	Purple
10	Dye grams/100 US gal min	0 793	0 50	1 00	0 60	0 793	0 50	1 00	0 985	0 60
11	Dye grams/100 US gal max	0 915	0 567	1 167	0 794	0 915	0 567	1 167	1 183	0 794
12	Freezing point max F	—76	—76	—76	—76	—76	—76	—76	—76	—76
13	Octane number lean min	80	91	100		80	91	100		
14	Performance number lean min	58 3	76	100	115	58 3	76	100	108	115
15	Octane number rich min		96			87	98			
16	Performance number rich min		87 5	130	145	68	87 5	130	135	145
17	Heat value Btu per lb min	18 700	18 700	18 700	18 700	18 700	18 700	18 700	18 800	18 800
18	Sulfur per cent by weight max	0 05	0 05	0 05	0 05	0.05	0 05	0 05	0 05	0 05
19	Inhibitors permitted	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes	Yes
20	Existent gum mg/100 ml max	3 0	3 0	3 0	3 0					
21	Copper dish res mg/100 ml max					5 0	5 0	5 0	5 0	5 0
22	Accelerated gum (5 hrs) mg/100 ml max					6 0	6 0	6 0	6 0	6 0
23	Accelerated gum (16 hrs) mg/100 ml max	6 0	6 0	6 0	6 0	10 0	10 0	10 0	10 0	10 0
24	Precipitate (5 hrs) mg/100 ml max					3 0	3 0	3 0	3 0	3 0
25	Precipitate (16 hrs) mg/100 ml max	2 0	2 0	2 0	2 0	4 0	4 0	4 0	4 0	4 0
26	Corrosion copper strip	None	None	None	None	None	None	None	None	None
27	Water tolerance permitted	None	None	None	None	None	None	None	None	None
28	Reid vapor press lb sq in max	7 0	7 0	7 0	7 0	7 0	7 0	7 0	7 0	7 0
29	Reid vapor press lb sq in min	5 5	5 5	5 5	5 5					. .

* Ethyl Corporation

† All purpose 80 octane automotive gasoline

its various components boil off at different temperatures, unlike water and alcohol (pure substances), which evaporate completely at one temperature.

Octane number (items 13 and 15, Table VI) is an index of the capacity of a gasoline to withstand high compression and high temperature without spontaneous burning or detonating. When detonation occurs, a reverberating shock wave is set up in the combustion chamber and an

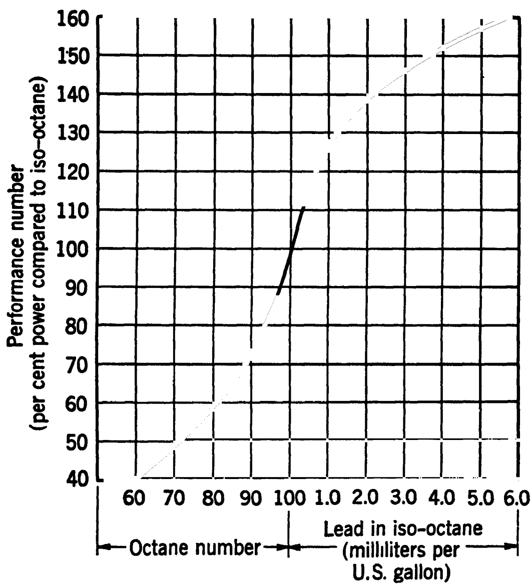


FIG. 43. Relationship between octane number and performance number.

audible ping results. , In severe cases it causes loss of power and possible stress failure of parts or overheating of the engine.

In order to compare gasolines a standardized single-cylinder engine is used in which all the variables such as speed, temperature, and mixture richness are held constant while different fuels are tried. *Iso-octane* (C_8H_{18}) is given a rating of 100, and *normal-* or *n-heptane* (C_7H_{16}) a rating of zero. Most other fuels fall between these two. By this method the octane scale was established, and any fuel which "knocks" or detonates as badly as *n*-heptane is called "zero octane," and fuel which performs as well as iso-octane is called "100 octane." A fuel that performs as well as 75 parts of iso-octane and 25 parts of *n*-heptane is described as "75 octane."

The addition of tetraethyl lead (TEL) improves the octane number of most gasolines. Adding TEL to iso-octane extends the scale or

"yardstick" of performance beyond that of 100 octane. It is customary to speak of iso-octane plus 1, 2, 3, or 4 milliliters, as the case may be, of tetraethyl lead per gallon when describing "super" fuel.

Performance number (items 14 and 16, Table VI) is a measure of the power that can be obtained from the gasoline by supercharging as compared to that which can be obtained from iso-octane by supercharging. Two comparisons are made, the first as a "lean" mixture and the second as a "rich" mixture. Thus a "115/130" performance number signifies that as a lean mixture this fuel will allow 15 per cent more power to be generated without knocking than can be obtained from a lean mixture of pure iso-octane; as a rich mixture this fuel will allow 30 per cent more power to be generated by supercharging than will a rich mixture of iso-octane. Consequently, 115/130 gasoline is better fuel than pure iso-octane but it may be no better than iso-octane plus 3 milliliters of TEL. Thus, performance number and the octane scale can be correlated well beyond the 100-octane point as shown in Fig. 43. For values below 100, such as 91/98, both numbers refer to the octane rating, the first number being the lean rating and the second the rich rating. Note that 60-octane fuel has a performance number of approximately 40, or only 40 per cent of the power can be generated with it as compared to iso-octane for the same mixture richness.

The primary factor limiting the power which can be developed in an Otto engine is detonation. It is not fully understood why, under like circumstances, one gasoline detonates and another does not. Neither is the action of tetraethyl lead fully understood. The most accepted explanation is that the shock wave of detonation, once started, travels across the combustion chamber at the speed of sound, the flame front moving with it or being generated by chain reaction. The presence of tetraethyl lead or ethyl fluid is thought to prevent chain reaction in certain hydrocarbons.

Ethyl fluid is the name given to a mixture of tetraethyl lead, Fig. 44(*a*), and ethylene dibromide, Fig. 44(*b*), or other bromine compounds. Dur-



(*a*) (*b*)

FIG. 44. (*a*) Tetraethyl lead; R = C_2H_5. (*b*) Ethylene dibromide.

ing combustion the bromides combine with the lead to form lead bromide, which has a low boiling point and is expelled from the cylinder with the exhaust gas. If pure lead were added to the fuel it would

form lead oxide, which would condense in the cylinder and cause spark plugs to foul.

The arrangement of the atoms in the hydrocarbon molecule is known to have a great effect on combustion in an engine cylinder. For example, consider heptane (C_7H_{16}), which is a saturated paraffin and is found in several forms. As a straight-chain or normal heptane, Fig. 45(*a*), it is one of the worst gasolines known for detonating. Another form of heptane called triptane, Fig. 45(*b*), is a super fuel having twice the performance number of iso-octane.

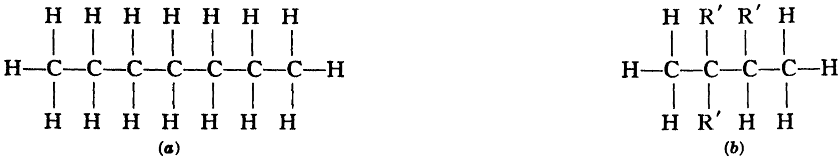


FIG. 45. (*a*) Normal heptane, C_7H_{16}. (*b*) Triptane, C_7H_{16}, or 2,2,3-trimethyl butane; R′ = methyl radical, or CH_3.

85. Fuel Oils. Essentially all fuel oils are derived from petroleum and consist of *residuals*, *distillates*, and *blends*. Residual oils are those which are not evaporated in the refining process and are, therefore, the lowest priced. They may contain impurities which will attack metal, such as that in turbine blades, at high temperatures. Being very viscous, or semisolid, at normal temperatures such oils usually require heating before a pump can be used; consequently, they are transported and stored in tanks having steam coils. Preheating is usually necessary to get proper control and mixing of oil and air before combustion.

Distillates and blended oils can be produced to meet, within limits, the requirements of such equipment as high-speed Diesel engines and domestic oil burners. Since heated storage is impractical in these applications, the oil must be fluid. A very clean fuel is required to prevent abrasion in a Diesel injection system. Likewise corrosive properties cannot be tolerated.

Kerosene is a petroleum distillate which is driven off at temperatures between 400 and 550 F. It is much less volatile than gasoline and has a low viscosity. It is widely used as a gas-turbine fuel.

The compositions of both crude petroleums and fuel oils, Table VII, do not show a wide range of variation; the same is true of their heating values per pound of oil. Fuel oils are standardized in five grades, which are designated by numbers; those listed as 1, 2, and 4 are used primarily in burners of domestic heating equipment; those classified as 5 and 6 are industrial fuel oils and are used in burners equipped for preheating. Owing largely to the differences in weights per gallon, the

heating value per gallon ranges from approximately 136,000 Btu for No. 1 oil to 152,000 Btu for No. 6 oil.

TABLE VII

ANALYSES OF TYPICAL AMERICAN OILS*

Location	Physical Properties				Chemical Properties				
	Gravity API at 60 F	Specific Gravity at 60 F	Flash Point, Degrees F	Burning Point, Degrees F	C, Carbon	H, Hydrogen	O, Oxygen	S, Sulfur	Calorific Value (Btu per pound)
California crude	16.3	0.957	172	210	86.58	11.61	0.74	0.82	18,613
Pennsylvania crude	39.8	0.826	...	.	82.00	14.80	3.20†	...	17,930
West Virginia crude	36.7	0.841	.	.	84.36	14.10	1.60†	. .	18,400
Texas crude	22.3	0.920	142	181	84.60	10.90	2.87†	1.63	19,060
Oklahoma crude	25.3	0.903	264	286	87.93	11.37	0.19	0.41	19,650
Texas fuel	21.3	0 926	216	240	83.26	12.41	3.83	0.50	19,481
Pennsylvania fuel	28 2	0.886		.	84.90	13.70	1.40†	...	19,200

* Stephen O. Andros, *Fuel Oil in Industry.*
† Includes nitrogen.

Table VIII lists some typical properties of furnace and Diesel engine fuels. The *carbon-residue* determination indicates the tendency to form carbon in the combustion zone with consequent clogging of the burner or nozzles. The *flash point* is significant as an index of fire hazard and possible ignition properties. It is desirable to have a low *sulfur percentage* since sulfur burned to completion forms sulfur dioxide, which may combine with the water resulting from the burning of hydrogen and produce sulfuric acid.

The oil industry expresses specific gravity in degrees, on the American Petroleum Institute (API) scale. *API gravity* indicates the weight of oil per gallon as well as the heating value per pound. The lighter oils have more energy per pound than heavy oils as indicated in Table V. Specific gravity and API gravity are related as follows:

$$\text{Specific gravity} = \frac{141.5}{131.5 + \text{degrees API}} \qquad (141)$$

The relation between specific gravity and the *higher heating value* (HHV) is given by Fig. 46. The relation between the higher heating value, Art. 88, and API gravity is expressed algebraically for fuel oil by the following empirical equation:[3]

$$\text{HHV} = 18{,}650 + 40(\text{API deg} - 10) \quad \text{Btu per lb} \qquad (142)$$

[3] Sherman-Kropf empirical equation. National Bureau of Standards, U. S., *Miscellaneous Publication* 97, "Thermal Properties of Petroleum Products."

TABLE VIII

Fuel Oil Specifications

Fuel	Cetane No	Viscosity SU Sec at 100 F	Water and Sediment per cent	Carbon Residue per cent	Ash per cent	Flash deg F	Sulfur per cent	ASTM Distillation deg F: Boiling Point	10 per cent	20 per cent	End Point	API Gravity	Pour Point deg F
Sun Oil medium furnace oil		28 34	0 05	0 10	0 0	125*	0 15†	355*	440†	620†	750†	34–39	5†
Sun Oil light Diesel oil		42†	0 10†	0 10†	0 05†	135*	0 15†	395	447	638	710	27–32	0
Phillips Pet No 2			0 05†			190			450†	620†		30–32	
No 3		70†	0 1			200			460†	620*		26–32	
Sinclair "Superflame"													
250	50	33	0	0 02	Trace	150	0 5	420	440	535	600	34–38	0
355	50	35	0	0 2	0	150	0 25	390–410	500	612	675	36–39	5
(Winter) 355	50	33	0	0 2	0	130	0 25	360	422	608	660	37–40	Below 0
Union Oil "Diesel"	56	41		0 05		200	0 65					32 5	0
Cities Service													
Mass No 2			0 05†	0 05		160	0 75		440†	620†	680†		0
Mass No 3		70	0 1†	0 15		200	0 75			620*			10
Tulsa Special Diesel No 3	49	35		0 05	0	185	0 18	312	440	560	626	36 5	−10
Tulsa Special Diesel No 1	55	35		Trace	0		0 073	303	410	581	692	39 6	Below 0
Tulsa Special Diesel No 2	57	44		0 038	Trace	250	0 12	473	534	622	663	34 9	20
Philadelphia Special Diesel No 3			0 1	0 15†		150*	0 75†			620–675		28–34	15–0
Philadelphia Special Diesel No 1			0 05	0 02†		110 150	0 5†		410†		600†	35 40	0–15
Philadelphia Special Diesel No 2			0 05	0 05†		125–140	0 5†		420†	600†	600*	32–38	0–15

* Minimum
† Maximum

The formula on page 103 may be used with reasonable accuracy for gasoline by replacing the first term with 18,320, and for kerosene by replacing the first term with 18,440.

In high-speed Diesel engines, if the fuel oil has poor ignition properties, it is slow to ignite on entering the cylinder and the charge may accumulate before any burning starts. When ignition is established

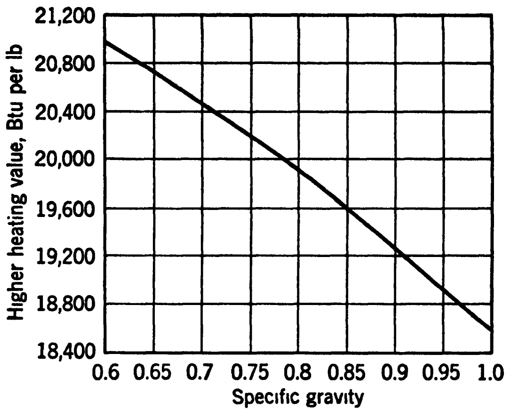
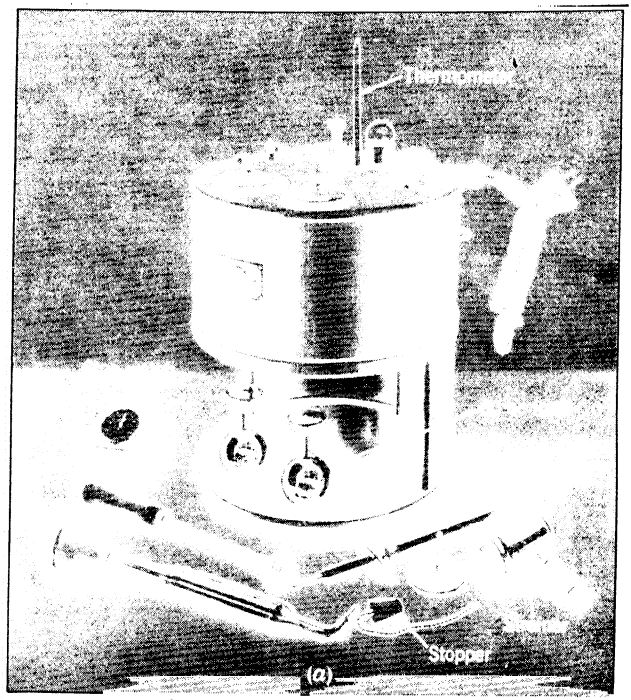


FIG. 46. Energy content vs specific gravity of fuel oil.

the accumulation is apt to burn with a suddenness that results in rough engine operation. Cetane ($C_{16}H_{34}$), a fuel easily ignited by compression, is used as a standard with a rating of 100, and alpha-methyl-naphthalene, which is difficult to ignite, is rated zero. Other fuel oils usually fall between these two extremes. The *cetane number* of a fuel represents the per cent by volume of cetane (*n*-hexadecane) in a blend of cetane and alpha-methyl-naphthalene that has the same ignition delay as the fuel in question. The comparison is made in a special test engine.

86. Viscosity Measurement. Figure 47 shows a laboratory type of Saybolt viscosimeter. By means of this apparatus the time required for a 60-cc sample of oil to flow through the calibrated outlet tube is called the Saybolt viscosity (in seconds) of the oil. In making the test the oil must be maintained at a definite temperature by a controlled water bath. The Universal Saybolt viscosimeter is used for oils requiring 32 to 1000 sec for passage through the tube. For oils of greater viscosity a tube of twice the diameter of the Universal is used and is called Saybolt Furol.

87. Alcohol. Alcohol is used rather widely as a fuel in countries having no petroleum resources, but having, instead, abundant plant



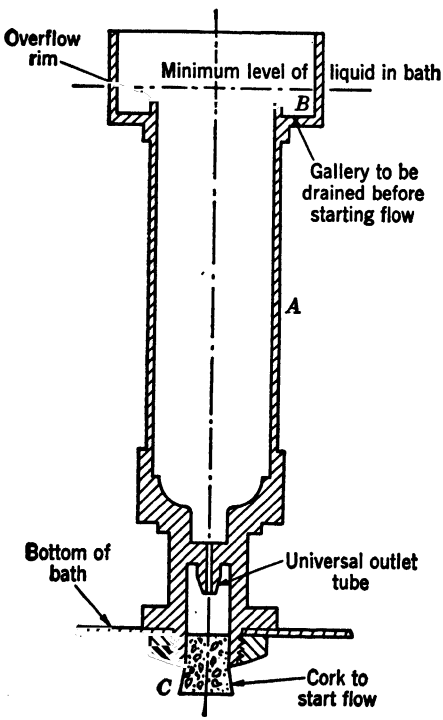


FIG. 47. (*a*) Aminco Saybolt viscosimeter. (*b*) Cross-sectional detail.

and vegetable material that can be processed. Cuba, Central America, and the Philippine Islands are examples. Four kinds of alcohol that can be used as fuel are (1) *ethyl* (*grain*), C_2H_6O; *methyl* (*wood*), CH_4O; (3) *butyl*, $C_4H_{10}O$; and *propyl*, C_3H_8O. Table V shows properties for ethyl and methyl alcohols. Alcohol has good anti-knock properties, its octane number being approximately 100.

88. Heating Value of Fuels. The energy liberated by the complete oxidation of a unit weight or volume of a fuel is its heating value. For solid and liquid fuels the heating value is expressed in Btu per pound and for gaseous fuels in Btu per standard cubic foot of gas measured at a temperature of 60 F and a pressure of 30 in. of mercury absolute.

Fuels which contain hydrogen have two heating values, the higher and the lower. The combustion of hydrogen produces water vapor which in the ordinary furnace escapes at the temperature of the chimney gases. The *lower heating value* is the heat liberated per pound of fuel after that necessary to vaporize the steam formed from the hydrogen has been deducted. The *higher heating value* of a fuel is the one indicated by a constant-volume calorimeter in which the steam produced is condensed and the heat of the vapor is recovered. The heating value of a fuel may be found by calculation as well as by the use of a calorimeter. Calorimeters are described in Arts. 89 and 90.

In the calculation of the heating value of coal from the ultimate analysis, only carbon, hydrogen, and sulfur are considered. Not all the hydrogen indicated by the ultimate analysis is available for combustion since a part is already in combination with oxygen in the form of inherent moisture. The available hydrogen is $[H - (O/8)]$ where H and O represent the proportionate parts[4] of hydrogen and oxygen, *by weight*, per pound of fuel. The energy liberated by the complete combustion of 1 lb of each of the following elements is carbon, 14,540 Btu; hydrogen, 62,000 Btu; and sulfur, 4050 Btu.

Dulong's formula for the heating value of a fuel based on the proportions *by weight* of the total carbon, the available hydrogen, and the sulfur is expressed by

$$\text{Btu per lb} = 14{,}540C + 62{,}000\left(H - \frac{O}{8}\right) + 4050S \qquad (143)$$

where C, H, O, and S are, respectively, weights of carbon, hydrogen, oxygen, and sulfur per pound of fuel. Dulong's formula gives an approximation of the higher heating value of a fuel.

The heating values calculated by Dulong's equation agree, in general, with those obtained with a calorimeter. Certain coals, such as Illinois

[4] The weight of hydrogen in water is one-eighth of the weight of the oxygen.

coals, may contain calcium carbonate, which may lead to an erroneous ash determination so that the percentages of oxygen and hydrogen are also incorrect. A discrepancy between the calculated and the calorimeter-determined heating value results.

89. Solid- and Liquid-Fuel Calorimeters. The cross section of a Parr oxygen bomb is shown in Fig. 48. It is made of noncorrosive material, and the essential parts of the bomb *F* are the cover *A*, the cover clamp nut *C*, the supports *DD*, the fuel tray *G*, the rubber gasket

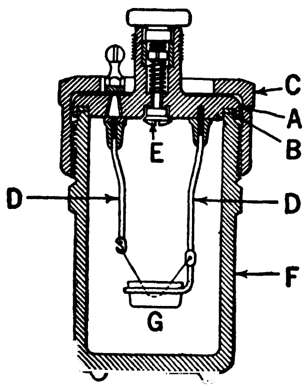


FIG. 48. Oxygen bomb details.

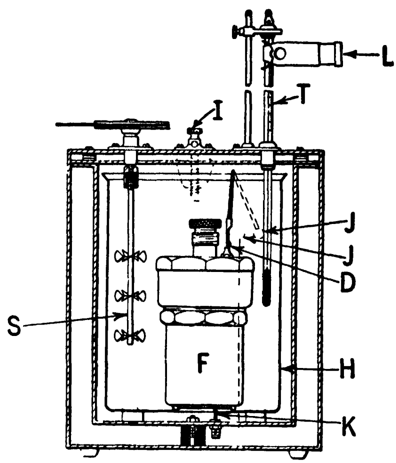


FIG. 49. Oxygen bomb calorimeter assembly.

B, and the check valve *E*. The supports *DD* serve as electrodes, and a measured amount of fine fuse wire is looped between them as shown so that it dips into the fuel held in the fuel tray *G*. Enough water is placed in the bomb to saturate the oxygen and air.

The assembled bomb *F* containing a carefully weighed amount, approximately 1 gram, of finely pulverized air-dried coal is filled with oxygen under 25 atmospheres pressure, and then placed in the calorimeter body as shown in Fig. 49. Part *H*, an oval-shaped metal container, holds exactly 2000 grams of distilled water which is kept in motion by the motor-driven stirrer *S*. The body of the calorimeter, including the cover and the bottom, is made of indurated fiber and has an air space in the walls. This body provides considerable resistance against the flow of heat to or from the surrounding atmosphere. Part *I* represents one of two clamps for holding the body cover in place. Low-voltage current is applied through the binding post *JJ* for ignition of the fuel

sample, electrical contact with the bomb fuse wire being made at *K* and *D*.

Temperature observations of the water in the container *H* are made by means of the thermometer *T* and the magnifying glass *L*. After equilibrium has been established, with the stirrer *S* in operation, the fuel charge is ignited, and temperature observations are continued until there is no further rise in temperature. The total temperature rise of the water in degrees F is multiplied by the so-called ***calorimeter constant.*** This constant is obtained from the sum of the grams of water divided by its specific heat (unity) and the grams of metal parts of the calorimeter divided by the mean specific heat of the metal. A simple expression can then be used for determining the heating value of a fuel.

$$\text{HHV} = \frac{\text{Calorimeter constant (grams)} \times \text{Temperature rise (deg F)}}{\text{Weight of fuel fired (grams)}} \qquad (144)$$

Since the fuel is weighed in grams on a chemical balance and the calorimeter constant is expressed as a gram equivalent of water, the heating value of the fuel is obtained directly in Btu per pound. The bomb calorimeter measures the higher heating value directly, if the bomb contains enough water before firing to saturate the oxygen. Any water formed from the combustion of hydrogen will therefore be condensed. The lower heating value can be obtained by subtracting from the higher heating value the latent heat of vaporization of the water formed from the combustion of the available hydrogen. Corrections are necessary for the heat of oxidation of the fuse wire and the heat liberated as the result of the formation of sulfuric and nitric acids within the bomb.

The procedure in the calorific determinations of liquid fuels is similar to that just described. When the liquid fuel is volatile it is necessary that it be weighed in a glass bulb, which is broken in the fuel tray just before the bomb is closed. This procedure prevents the loss of volatile constituents of the fuel during the weighing operation.

90. Gas Calorimetry. The Sargent calorimeter shown by Fig. 50 is typical. The gas, for which a heating-value determination is to be made, is passed through a calibrated meter *D*, and its pressure and temperature are noted. A pressure regulator in the meter outlet line maintains the pressure of the gas nearly uniform as it flows to a carefully adjusted Bunsen burner *B* located within the calorimeter body. The calorimeter body *C* is a vertical cylindrical chamber having a water jacket through which water is allowed to flow under a constant head. Calibrated thermometers *TT* are placed in the inlet and outlet water connections. The heat produced by the gas is determined by accurately

weighing the water passing through the calorimeter while a given quantity of gas is burned. The rate of water flow is adjusted until the water enters and leaves within a few degrees of room temperature. Likewise the exit gases from the burner are cooled to room temperature within

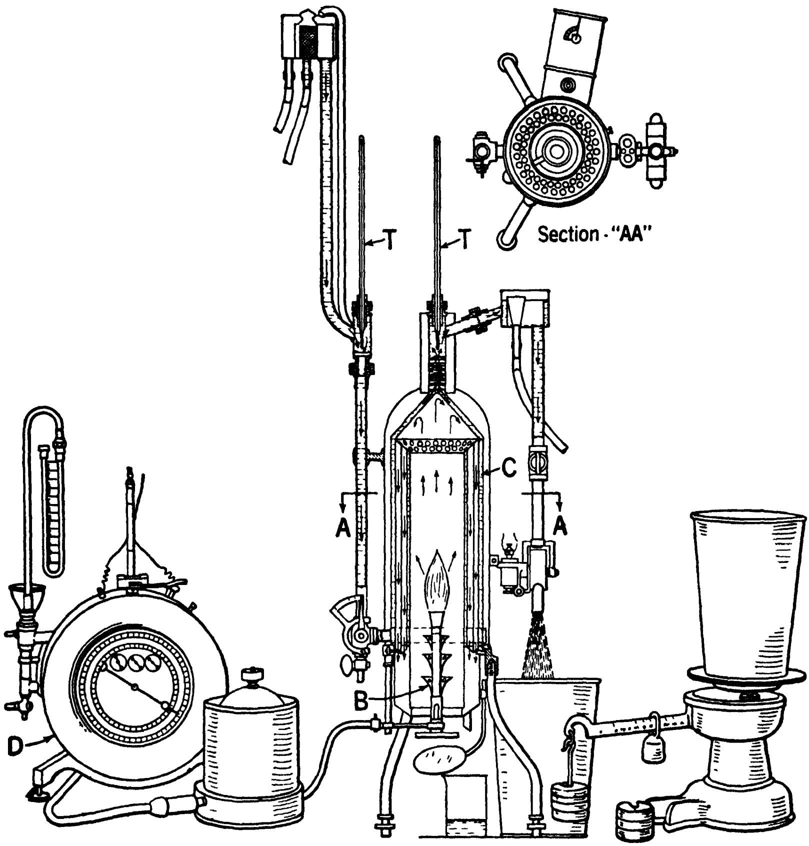


FIG. 50. Sargent gas calorimeter.

the calorimeter; thus all the energy from the fuel is absorbed by the water and there is practically no radiation loss. Frequent readings of the inlet and outlet water temperatures are taken during this period, and the average temperature rise of the water is determined. The product of the temperature rise and the weight of water passing through the calorimeter, divided by the cubic feet of gas burned, gives the heating value of the fuel per cubic foot as measured. This result may be corrected to the heating value per cubic foot at 60 F temperature and

30 in. of mercury absolute pressure. The heating value thus determined is the lower heating value *if no condensation occurs.* If temperatures and humidity are controlled to give complete condensation, then the higher heating value is obtained directly. In most gas calorimeters not all water is condensed and corrections must be made. The difference between the higher and lower heating values is the product of the weight of water vapor formed per cubic foot of fuel burned and its change in enthalpy.

91. Combustion Chemistry. Combustion is synonymous with oxidation and is the union of oxygen with a combustible material. The rates of combustion vary widely, slow combustion being called ***burning*** and very rapid being called *detonation.* From basic chemistry the union of carbon and oxygen is indicated as follows:

$$C + O_2 \rightarrow CO_2 \tag{145}$$

$$1 + 1 \rightarrow 1 \quad \text{mole basis}$$

$$12 + 32 \rightarrow 44 \quad \text{weight basis} \tag{146}$$

The above equations show that 1 molecular weight of carbon combines with 1 molecular weight of oxygen to produce 1 molecular weight of carbon dioxide. A mole is M pounds, where M is the molecular weight. Although the combining quantities can be expressed as pounds, it is simpler to make all computations on a mole basis and convert to pounds or cubic feet only when necessary.

Complete combustion occurs when the fuel is fully oxidized and all the energy is released. Thus equation 145 shows complete oxidation of carbon. *Incomplete combustion* may result from (1) insufficient oxygen, (2) poor mixing of fuel and oxygen, or (3) a temperature too low to support combustion. Equations 147 and 148 show incomplete combustion since carbon monoxide and free hydrogen remain in the respective products.

$$C + \tfrac{1}{2}O_2 \rightarrow CO \tag{147}$$

$$2H_2 + \tfrac{1}{2}O_2 \rightarrow H_2O + H_2 \tag{148}$$

The oxygen for combustion is always taken from the air, and much nitrogen is brought along with it since the two cannot be easily separated. Bottled oxygen is used for special operations such as cutting torches in which the diluting effects of nitrogen cannot be tolerated. It is therefore necessary to review the properties of air as a source of oxygen.

Table IX shows the percentages of the major components of dry air. There are traces of many other gases. Each gas exists at its own par-

TABLE IX

COMPOSITION OF DRY AIR

	Per Cent by Volume, moles	Per Cent by Weight	Molecular Weight, lb per mole
Oxygen	20.99	23.19	32.00
Nitrogen	78.03	75.47	28.016
Argon	0.94	1.30	39.944
Carbon dioxide	0.03	0.04	44.003
Hydrogen	0.01	0.00	2.016
Dry air	100.00	100.00	28.967

tial pressure throughout the atmosphere as though the others were not present (Dalton's law). For slide-rule computations assume air to be 21 per cent oxygen and 79 per cent nitrogen by volume, 23.2 per cent oxygen and 76.8 per cent nitrogen by weight. All the constituents except oxygen may be treated as inert gas for which the molecular weight is 28.2 and that of pure nitrogen is 28.0.

The water-vapor content of air varies widely with temperature and per cent saturation. Saturated air at 70 F contains 2.46 per cent moisture by volume. Air at 70 F and 50 per cent saturation has $0.50 \times 2.46 = 1.23$ per cent moisture by volume, which means that moisture by volume is in direct proportion to the per cent saturation. Saturation at 212 F indicates 100 per cent water vapor. The amount of water vapor in air can be calculated with sufficient accuracy from a psychrometric chart if the wet- and dry-bulb temperatures are known. In furnace operation the loss due to moisture is calculated as part of the energy balance, Art. 111.

The molar ratio of nitrogen to oxygen in the air is the same as their volumetric ratio, since both are gases and at the same temperature; thus

$$\frac{\text{moles } N_2}{\text{moles } O_2} = \frac{79 \text{ per cent}}{21 \text{ per cent}} = 3.76+ \tag{149}$$

The above ratio shows that for each mole of oxygen in the air there are 3.76 moles of nitrogen. Thus, 3.76 volumes of nitrogen are drawn in through the air filter, the carburetor, the intake valves, and the cylinder of an internal-combustion engine for each volume of oxygen obtained.

The *combustion of carbon* in air is represented by the equation

$$\begin{array}{llllll} C + & O_2 & + 3.76N_2 & \rightarrow CO_2 & + 3.76N_2 & \quad (150) \\ 1 + & 1 & + 3.76 & \rightarrow 1 & + 3.76 & \text{mole basis} \\ 12 + & 32 & + 3.76(28.2) & \rightarrow 44 & + 3.76(28.2) & \text{weight basis} \end{array}$$

Dividing by 12,

$$\underset{\text{Fuel}}{1} + \underset{\text{Air}}{2.667 + 8.84} \rightarrow \underset{\text{Products}}{3.667 + 8.84} \quad \text{weight basis} \qquad (151)$$

Therefore, the air required is

$$(2.667 + 8.84)/1 = 11.5 \text{ lb per lb of carbon} \qquad (152)$$

The *combustion of hydrogen* in air is represented by the equation

$$\begin{array}{llllll} 2H_2 + O_2 + & 3.76N_2 & \rightarrow 2H_2O + & 3.76N_2 & & \quad (153) \\ 2 + 1 + & 3.76 & \rightarrow 2 + & 3.76 & \text{mole basis} \\ 2(2) + 32 + & 3.76(28.2) & \rightarrow 2(18) + & 3.76(28.2) & \text{weight basis} \end{array}$$

Dividing by 4,

$$\underset{\text{Fuel}}{1} + \underset{\text{Air}}{8 + 26.5} \rightarrow \underset{\text{Products}}{9 + 26.5} \quad \text{weight basis} \qquad (154)$$

Therefore, the air required is

$$(8 + 26.5)/1 = 34.5 \text{ lb per lb of hydrogen} \qquad (155)$$

The *combustion of sulfur* in air gives

$$\begin{array}{llllll} S + O_2 + 3.76N_2 & \rightarrow SO_2 + 3.76N_2 & & \quad (156) \\ 1 + 1 + 3.76 & \rightarrow 1 + 3.76 & \text{mole basis} \\ 32 + 32 + 3.76(28.2) & \rightarrow 64 + 3.76(28.2) & \text{weight basis} \end{array}$$

Dividing by 32,

$$\underset{\text{Fuel}}{1} + \underset{\text{Air}}{1 + 3.32} \rightarrow \underset{\text{Products}}{2 + 3.32} \quad \text{weight basis} \qquad (157)$$

Therefore, the air required per pound of sulfur is

$$(1 + 3.32)/1 = 4.32 \text{ lb} \qquad (158)$$

To illustrate the *combustion of hydrocarbons*, consider typical gasoline (C_8H_{18}).

$$\begin{array}{llllll} C_8H_{18} + (12.5)O_2 + 12.5(3.76N_2) & \rightarrow 8CO_2 + 9H_2O + 12.5(3.76N_2) & & \quad (159) \\ 1 + 12.5 + 47 & \rightarrow 8 + 9 + 47 & \text{mole basis} \\ 114 + 400 + 1325 & \rightarrow 352 + 162 + 1325 & \text{weight basis} \end{array}$$

Dividing by 114,

$$\underset{\text{Fuel}}{1} + \underbrace{3.51 + 11.62}_{\text{Air}} \rightarrow \underbrace{3.09 + 1.42 + 11.62}_{\text{Products}} \quad \text{weight basis} \tag{160}$$

Thus, for each pound of this gasoline, (3.51 + 11.62)/1 or 15.13 lb of air are required for combustion, the air consisting of 3.51 lb of oxygen and 11.62 lb of nitrogen. The *air-fuel ratio* is therefore 15.13 to 1, and the *fuel-air ratio* is 1 to 15.13 or 0.066. These expressions are widely used as an indication of the *richness* or *leanness* of a fuel-air mixture. Equation 159, for example, illustrates a *chemically correct, ideal,* or *perfect mixture.* The chemically correct proportions will not produce complete combustion since, by the law of probability, not every one of the tremendous multitude of fuel molecules will find an oxygen molecule with which to combine. To be sure that all the fuel is oxidized it is necessary to use a mixture that is on the lean side, i.e., that has some *excess air.*

With a deficiency of air, heavy losses in the form of carbon monoxide, free hydrogen, and distilled hydrocarbons can occur. In furnace operation the object is to use the minimum of excess air and still get complete combustion. The amount of excess air necessary in furnace operation depends on (1) how much time is available before the rising gases reach the cool tube bank and become chilled below the ignition point, (2) at what temperature the mixture exists, and (3) how intimately the fuel and air can be mixed. These three primary factors are frequently summarized as "time, temperature, and turbulence." If a fuel can be highly dispersed and thoroughly mixed with the air, it can be burned with a small amount of excess air.

Example. Typical fuel oil, $C_{16}H_{32}$, is burned with the correct amount of air. Find (1) the combustion equation, (2) the ideal air-fuel ratio, (3) the percentage of CO_2 by volume in the dry stack gases, and (4) the percentage of N_2 by volume in the dry stack gases.

Solution.

(1) $$C_{16}H_{32} + (24)O_2 + 3.76(24N_2) \rightarrow 16CO_2 + 16H_2O + 3.76(24N_2)$$

$$\underset{\text{Fuel}}{1} + \underbrace{24 + 90.24}_{\text{Air}} \rightarrow \underbrace{16 + 16 + 90.24}_{\text{Products}} \quad \text{mole basis}$$

(2) $$\frac{A}{F} = \frac{\text{lb air}}{\text{lb fuel}} = \frac{24(32) + 90.24(28)}{16(12) + 32(1)} = 14.68 \text{ lb}$$

(3) To obtain per cent CO_2 by volume in the dry stack gases,

$$\frac{16}{16 + 90.24} = 0.1505 \quad \text{or} \quad 15.05 \text{ per cent } CO_2$$

(4) $$\text{Per cent } N_2 = 100 - 15.05 = 84.95$$

Note that 15.05 per cent by volume of CO_2 is the maximum that can be obtained and therefore represents perfect combustion with no excess air.

Example. Typical gasoline, C_8H_{18}, is burned with 20 per cent excess air by weight. Find (1) the combustion equation, (2) the air-fuel ratio, pound per pound, (3) the per cent CO_2 by volume in the dry exhaust gases, (4) pounds of water vapor formed per pound of fuel, (5) volume of dry exhaust gas per pound of fuel if $t = 60$ F and $p = 14.7$ psia, and (6) partial pressure of the water vapor in the exhaust.

Solution. For correct air-fuel ratio

$$C_8H_{18} + (12.5)O_2 + 12.5(3.76N_2) \rightarrow 8CO_2 + 9H_2O + 12.5(3.76N_2)$$

With 20 per cent excess air

(1) $$\underset{\text{Fuel}}{C_8H_{18}} + \underset{\text{Air}}{1.2(12.5)O_2 + 1.2(12.5)3.76N_2} \rightarrow \underset{\text{Pr ducts}}{8CO_2 + 9H_2O + 1.2(12.5)3.76N_2 + (2.5)O_2}$$

(2) $$\frac{A}{F} = \frac{1.2(12.5)32 + 1.2(12.5)(3.76)28.2}{12(8) + 18} = 18.05 \text{ lb per lb}$$

(3) To obtain per cent CO_2 in dry gases,

$$\frac{8}{8 + 1.2(12.5)3.76 + 2.5} = 0.1195, \quad \text{or} \quad 11.95 \text{ per cent}$$

(4) $$\frac{\text{lb } H_2O}{\text{lb fuel}} = \frac{9(2 + 16)}{12(8) + 18} = 1.42 \text{ lb per lb}$$

(5) $$PV = NBT \quad \text{or} \quad \frac{V}{m_f} = \frac{NBT}{P \times m_f} \quad (m_f = \text{weight of fuel fired, lb})$$

$$\frac{V}{m_f} = \frac{[8 + 1.2(12.5)3.76 + 2.5](1545)520}{14.7(144)114} = 223 \text{ cu ft per lb of fuel}$$

(6) $$\frac{P'_{H_2O}}{N_{H_2O}} = \frac{P_T}{N_T} \qquad P'_{H_2O} = \frac{9(14.7)}{8 + 9 + 1.2(12.5)3.76} = 1.805 \text{ psia}$$

Note that in the *dry products* the water has been condensed.

92. Theoretical Air for Combustion. The theoretical weight of air required to burn a pound of fuel for which the chemical analysis is known, such as coal, is

$$m_{ta} = 11.5C + 34.5\left(H - \frac{O}{8}\right) + 4.32S \tag{161}$$

where C, H, O, and S represent the weights of carbon, hydrogen, oxygen, and sulfur, respectively, per pound of fuel.

The net weight of hydrogen available for combustion is assumed to

be all that which is not already in combination with oxygen as water. The chemist's analysis will report the remaining moisture in coal as hydrogen and oxygen. It is then assumed that all the O_2 is in combination with H_2. The combining ratio of H_2 to O_2 is 1 to 8 by weight; therefore the net H_2 left for further oxidation is (H − O/8).

Example. Find the theoretical weight of air required to burn 1 lb of Clay county, Missouri, coal, Table III, for which the ultimate analysis in per cent, by weight, as received, is S = 2.97; C = 58.62; H_2 = 4.05; O_2 = 7.54; N_2 = 1.08; and moisture = 15.83.

Solution.

$$m_{ta} = 11.5C + 34.5\left(H - \frac{O}{8}\right) + 4.32S$$

$$= (11.5 \times 0.5862) + 34.5\left(0.0405 - \frac{0.0754}{8}\right) + (4.32 \times 0.0297)$$

$$= 7.94 \text{ lb per lb}$$

Example. The analysis of a natural gas showed the following percentages by volume: ethane, 9.0; methane, 90.0; carbon dioxide, 0.2; and nitrogen, 0.8. Find the theoretical volume of air required per cubic foot of gas if the gas and air are at a temperature of 60 F and a pressure of 30 in. of mercury.

Solution. In this gas the ethane (C_2H_6) and the methane (CH_4) are the only combustible materials.

$$C_2H_6 + (3.5)O_2 + 3.5(3.76N_2) \rightarrow 2CO_2 + 3H_2O + 3.5(3.76N_2)$$

$$\underset{\text{Fuel}}{1} + \underbrace{3.5 + 13.16}_{\text{Air}} \rightarrow \underbrace{2 + 3 + 13.16}_{\text{Products}}$$

$$16.66/1 = \text{molar or volume ratio, air to fuel}$$

$$0.09(16.66) = 1.5 \text{ cu ft of air for } C_2H_6$$

$$CH_4 + (2)O_2 + 2(3.76N_2) \rightarrow CO_2 + 2H_2O + 2(3.76N_2)$$

$$\underset{\text{Fuel}}{1} + \underbrace{2 + 2(3.76)}_{\text{Air}} \rightarrow \underbrace{1 + 2 + 2(3.76)}_{\text{Products}}$$

$$9.52/1 = \text{molar or volume ratio, air to fuel}$$

$$0.90(9.52) = 8.568 \text{ cu ft of air for } CH_4$$

$$\text{Total air per cubic foot of gas} = 1.5 + 8.568 = 10.068 \text{ cu ft}$$

93. Combustion Products. The exhaust gases from engines and furnaces are called products of combustion. The composition of the products depends on the kind of fuel used, the air-fuel ratio, and the conditions of combustion. The products usually contain carbon dioxide, carbon monoxide, oxygen, nitrogen, free carbon, fly ash, steam, sulfur dioxide, and unburned hydrocarbons. Most of these products are invisible. Carbon monoxide in the exhaust can cause large energy

losses. Usually the appearance of heavy gaseous smoke is indicative of low-temperature operation or insufficient oxygen.

94. Analysis of Exhaust Products. An analysis of exhaust products by means of the common Orsat apparatus, Art. 95, will yield information from which the air-fuel ratio may be calculated. Also the completeness of combustion may be determined, and this information is vital to efficient operation of a furnace or engine. By means of an Orsat analysis of exhaust gases, modern garages adjust the carburetor air-fuel ratio. Likewise steam power plants continually record by means of an Orsat the CO_2 concentration in the exhaust products to check efficient operation. Maximum CO_2 concentration is the objective in all external-combustion plants.

Actual measurement of fuel and air flow may be made when extreme accuracy is required, but air-flow measurements necessitate extensive facilities and are expensive. The simplest method is to obtain a sample of the products of combustion and determine the percentage by volume of each component gas. Since these percentages are determined under conditions of constant pressure, temperature, and saturation, equal volumes of any of the various gases will contain the same number of molecules (Avogadro's law). Since nitrogen is inert the moles of nitrogen appearing in the exhaust products must also appear in the fuel-air mixture. This nitrogen serves as an index to the quantity of air entering because 3.76 moles of nitrogen enter with each mole of oxygen from the air. Likewise the carbon that appears in the exhaust products must be accounted for in the fuel.

In analyzing the products the water vapor is condensed to liquid and becomes part of the pumping fluid in the Orsat, and therefore does not enter into the analysis. The analysis is equivalent to one made on dry gases.

Example. An analysis of exhaust gases shows the products to consist of the following percentages by volume: $CO_2 = 12.1$, $O_2 = 0.3$, $CO = 3.3$, $H_2 = 1.3$, $CH_4 = 0.3$, $N_2 = 82.7$. Assuming the fuel to be a pure hydrocarbon of the form C_xH_y, (1) find the values for x and y, (2) balance the combustion equation, and (3) determine the air-fuel ratio.

Solution.

(1) $$C_xH_y + aO_2 + 82.7N_2 \rightarrow 12.1CO_2 + (0.3)O_2 + 3.3CO + 1.3H_2 + (0.3)CH_4 + 82.7N_2 + bH_2O$$

where a = moles of oxygen (in air supplied).

b = moles of water vapor (in products).

$$a = \frac{82.7}{3.76} = 22$$

$$x = 12.1 + 3.3 + 0.3 = 15.7$$

$$O_2 = 12.1 + 0.3 + \frac{3.3}{2} = 14.05 \text{ (in dry gases)}$$

$$b = (22 - 14.05)2 = 15.9$$
$$H_2 = 1.3 + (2 \times 0.3) + 15.9 = 17.8$$
$$y = 2 \times H_2 = 35.6$$

(2) The combustion equation is, therefore,

$$C_{15.7}H_{35.6} + (22)O_2 + 82.7N_2 \rightarrow 12.1CO_2 + (0.3)O_2 + 3.3CO + 1.3H_2 + 0.3CH_4 + 82.7N_2 + 15.9H_2O$$

(3) $$\text{Air-fuel ratio} = \frac{(22 \times 32) + (82.7 \times 28.2)}{(15.7 \times 12) + (35.6 \times 1)} = 13.5 \text{ lb per lb}$$

Although x and y have been determined, the fuel composition is not necessarily defined. Actually more or less than 1 mole of fuel may be involved such that x and y would reduce to whole numbers when multiplied by a constant, $z(C_xH_y)$. The proportion of hydrogen to carbon, called the hydrogen-carbon ratio, is more significant and can now be determined as H/C, or $y/x = 35.6/15.7 = 2.26$, or $CH_{2.26}$ may now be written. For all midwest oils and gasolines, 2.12 is a representative hydrogen-carbon ratio.

The analysis of any gas should be made of a typical sample. Great care should be taken in drawing a sample so that it is not diluted with air. Also the sample should be typical of the gases being produced at the time of its withdrawal. Where high temperatures are involved, metal sampling tubes are unsuitable, as they may be oxidized by the oxygen of the flue gases drawn and thus give an erroneous indication of the oxygen content. Fused-quartz tubes are much better than iron if they are to be heated to a red color.

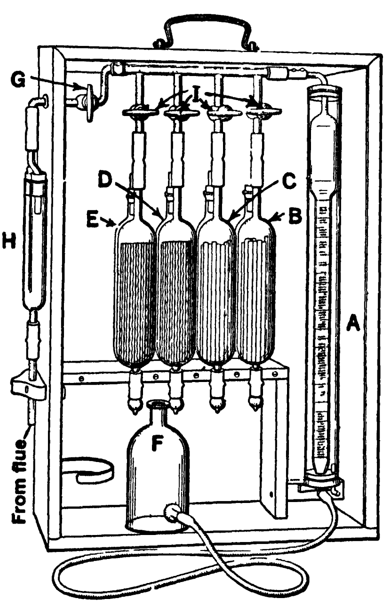


FIG. 51. Orsat apparatus.

95. Flue-Gas Analyzers. A typical Orsat apparatus for gas analysis is shown in Fig. 51. The essential parts are the measuring burette, A; the leveling bottle, F; the gas cleaner, H; and the absorption pipettes, B, C, D, and E, which are interconnected by means of a manifold having cocks I. Each of the absorption pipettes is fitted with a number

of small glass tubes which give a great amount of surface, wetted by the absorbing reagents, exposed to the gas under analysis. The measuring burette is water jacketed to prevent temperature and density changes of the gas. Carbon dioxide is absorbed in pipette *B*, which is partially filled with caustic potash, KOH; oxygen is taken out in pipette *C*, which contains an alkaline solution of pyrogallic acid; and carbon monoxide is absorbed by an acid solution of cuprous chloride in pipettes *D* and *E*. These pipettes should contain some metallic copper to keep the solution energized. The absorptions must occur in the order indicated, and the nitrogen is obtained by difference. The reagents should be fresh and kept protected from the air. All connections between the various parts of the apparatus must be leakproof.

96. Actual Weight of Combustion Air. Article 94 illustrates one method of treating exhaust analysis which is applicable for oil and gas, particularly if the chemical formula is known. In another method the analysis of the flue gases formed, the ultimate analysis of the fuel burned, and the molecular weights of the principal constituents of the flue gases provide data for calculating the actual amount of air used per pound of fuel. The percentages of CO_2, O_2, CO, and N_2 of the flue gases are to be taken by volume; the carbon content in the fuel is by weight.

If 100 mole volumes of flue gases are considered, the percentages of the flue-gas analysis represent the mole volumes of each constituent present. The weight of nitrogen is $28 \times N_2$, and the weight of carbon appearing as CO_2 and CO is $12(CO_2 + CO)$. Therefore, the weight of nitrogen per pound of carbon is

$$\frac{28N_2}{12(CO_2 + CO)}$$

As 1 lb of air contains 0.769 lb of nitrogen, the weight of air, m_c, per pound of carbon, is

$$m_c = \frac{28N_2}{12(CO_2 + CO) \times 0.769} \tag{162}$$

There is likelihood of unburned carbon appearing in the ash and refuse from solid fuels. Under such conditions, calculations involving the use of flue-gas analyses should be based on the carbon actually burned per pound of fuel. The actual carbon consumed per pound is

$$C_1 = \frac{m_f C_f - m_r C_r}{m_f \times 100} \tag{163}$$

where C_1 = carbon burned per pound of fuel, lb.
m_f = weight of fuel fired, lb.
C_f = carbon of the fuel from ultimate analysis, per cent.
m_r = weight of refuse from m_f lb of fuel, lb.
C_r = carbon content of the refuse, per cent.

If C_r has not been determined by analysis, it may be found by

$$C_r = \frac{100m_r}{m_f} - A \tag{164}$$

where A = per cent ash as obtained from ultimate analysis. Using the product of m_c and C_1,

$$m_{aa} = \frac{28N_2}{12(CO_2 + CO) \times 0.769} \times \frac{m_fC_f - m_rC_r}{m_f \times 100} \tag{165}$$

where m_{aa} = the actual weight of dry air used per pound of fuel fired, lb.

When account is taken of the nitrogen in coal, its percentage should be divided by 76.9, and the weight of air equivalent should be deducted from m_{aa}. This correction represents the air equivalent of the nitrogen content of the flue gases produced by the nitrogen in the fuel.

97. Weight of Excess Air. Excess air is generally expressed as a percentage of the theoretical requirements

$$E_a = \frac{m_{aa} - m_{ta}}{m_{ta}} \times 100 \tag{166}$$

where E_a = percentage of excess air.
m_{aa} = actual air used per pound of fuel, lb.
m_{ta} = theoretical air required per pound of fuel, lb.

The percentage of excess air required for good combustion may range from 10 to 60. Often, in burning solid fuels, holes in the fire, thin fires, strong draft, or leaky boiler and furnace settings may cause the actual percentage of excess air to be large. Improper adjustments of oil or gas burners may allow a great volume of excess air to pass through the furnace.

98. Weight of Dry Gases per Pound of Fuel. On the basis of reasoning identical with that used to express the actual air used per pound of fuel, the weight of the dry gases formed per pound of carbon is

$$m_g = \frac{44CO_2 + (32)O_2 + 28(CO + N_2)}{12(CO_2 + CO)} \tag{167}$$

The symbols CO_2, O_2, CO, and N all represent percentages by volume as determined by the flue-gas analysis.

Equation 167 may be simplified as

$$m_g = \frac{44CO_2 + (32)O_2 + 28[(100 - CO_2 - O_2 - N_2) + N_2]}{12(CO_2 + CO)}$$

$$m_g = \frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)} \tag{168}$$

The actual weight of dry gas formed per pound of fuel is found by $C_1 \times m_g$.

$$m_{dg} = \left[\frac{4CO_2 + O_2 + 700}{3(CO_2 + CO)}\right]\left[\frac{m_f C_f - m_r C_r}{m_f \times 100}\right] \tag{169}$$

The actual weight of *dry air supplied* per pound of fuel as fired may be obtained by use of equation 170. Such a calculation involves the deduction of the weight of carbon C_1 actually burned per pound of fuel from the weight of the dry gases and the addition of the oxygen necessary for the combustion of the hydrogen. The nitrogen from the air used for the burning of the hydrogen appears in the total nitrogen content of the flue gases.

$$m_{aa} = m_{dg} - C_1 + 8\left(H - \frac{O}{8}\right) \tag{170}$$

In equation 170, C_1 is determined as indicated by equation 163, and H and O are the fuel percentages each divided by 100. Calculations made by means of equation 170 will approximately check those made by means of equation 165.

PROBLEMS

1. A 7-lb sample of coal, as received, weighs 6.7 lb after air drying. When oven dried, 1.20 grams of the air-dried coal are reduced to a weight of 1.17 grams. Find the total percentage of moisture in the original coal. 6.8 per cent, *Ans.*

2. A moist coal loses 5 per cent of its original weight when air dried. A quantity of the air-dried coal is oven dried with a loss of 7 per cent of the weight of the air-dried material. What is the total moisture of the original coal, expressed as a percentage?

3. Express the ultimate analysis of Franklin County, Illinois, bituminous coal on both a dry and a moisture-and-ash-free basis.

4. Resolve the proximate analysis of Crawford County, Kansas, bituminous coal to both a dry and a moisture-and-ash-free basis.

5. Resolve the proximate analysis of Lackawanna County, Pennsylvania, anthracite coal to both a dry and a moisture-and-ash-free basis.

6. Compute by means of Dulong's equation the heating value of 1 lb of Franklin County, Illinois, bituminous coal as received and on a dry basis.

As rec'd: 11,370 Btu per lb; dry: 12,850 Btu per lb; *Ans.*

7. Find by use of Dulong's equation the heating value of 1 lb of Clay County, Missouri, bituminous coal in a dry condition and also per pound of combustible.

8. Calculate the heating value of 1 lb of West Virginia coal having the following analysis, percentage by weight: C, 78.71; H_2, 4.45; O_2, 4.33; N_2, 1.31; S, 0.68; moisture, 2.52; ash, 8.00.

9. The ultimate analysis of a coal, percentage by weight, shows C, 70.75; H_2, 4.62; O_2, 9.94; S, 0.54; moisture, 9.72; and ash, 4.43. When 19,435 lb of this coal are fired in a furnace, 981 lb of ash and refuse are recovered, of which 14 per cent is found to be carbon. An exhaust-gas analysis yields the following data, percentage by volume: CO_2, 11.0; O_2, 6.0; CO, 0.5. What is the percentage of excess air used?

64.0 per cent, *Ans.*

10. Coal with an ultimate analysis, percentage by weight, of C = 68.6, H_2 = 6.1, O_2 = 8.2, N_2 = 1.7, S = 3.1, ash = 12.30, is fired producing gases having the following analysis, percentage by volume: CO_2 = 13.33, O_2 = 6.16, CO = 0.0. Temperature of the air supplied is 80 F; temperature of gases leaving the furnace is 325 F; specific heat of the stack gases, c_p = 0.25 Btu per lb; unburned carbon is 0.02 lb per lb of coal fired. Find: (*a*) theoretical air, pounds per pound; (*b*) actual air used, pounds per pound; and (*c*) heat loss due to dry gases, Btu per pound of coal.

11. A coal having 61.7 per cent carbon requires theoretically 10.5 lb of air per lb for complete combustion. During the combustion process 2 per cent of the carbon in the coal is lost in the refuse. Find the percentage of excess air used if the flue gas analysis, percentage by volume, is CO_2 = 10.1, CO = 0.0, O_2 = 8.3, and N_2 = 81.6.

12. A fuel oil has an API gravity of 32 degrees at 60 F. Compute the energy per gallon of the oil. 141,000 Btu per gal, *Ans.*

13. Calculate the higher heating value at 77 F and 14.696 psia of 1 lb of a natural gas of the following composition, percentage by volume: CH_4, 94.3; C_2H_6, 4.2; CO_2, 1.5.

14. A gas storage tank of 25,000 cu ft volume contains gas at a pressure of 25 psia and a temperature of 40 F. The higher heating value of the gas is 600 Btu per cu ft at 60 F and 14.696 psia. Find the heating value of this gas, per cubic foot of tank volume.

15. When 1.2 grams of air-dried coal are burned in an oxygen-bomb calorimeter the temperature rise of 2000 grams of water is 6.52 F, of which 0.35 F is due to the fuse wire and acid formation. The water equivalent of the metal parts of the calorimeter is 450 grams. Find the heating value of the dry coal.

16. Natural gas, measured at a temperature of 80 F and a pressure of 5 in. water gage (temperature at 60 F), is burned in a gas calorimeter, giving the following data: temperature of the inlet water, 73.1 F; temperature of the outlet water, 88.1 F; weight of water used, 13.2 lb; gas used, 0.3 cu ft; and barometric pressure, 28 in. of mercury. Find the higher heating value of the gas, per cubic foot, at 60 F and 14.696 psia, assuming all water vapor formed is condensed. 724 Btu per cu ft, *Ans.*

17. Calculate the theoretical volume of dry air required for the complete combustion of 1 lb of Fayette County, Pennsylvania, coal as received, when the air temperature is 70 F and the barometric pressure is 29.92 in. of mercury.

18. Determine the theoretical weight of air required per pound for burning Franklin County, Illinois, coal as received.

19. Determine the theoretical air-fuel ratio required for $C_{12}H_{16}$.

13.75 lb per lb, *Ans.*

20. C_8H_{18} is burned with 10 per cent excess air. Assuming complete combustion, find (*a*) air-fuel ratio, pounds per pound; (*b*) per cent CO_2 as read by an Orsat; (*c*) volume of air required per pound of fuel, air at 14 psia and 100 F; and (*d*) the weight of dry gases formed per pound of oil.

21. C_7H_{16} is burned with the chemically correct weight of air. Assuming complete combustion, find (*a*) the combustion equation, (*b*) per cent of CO_2 in the dry exhaust gases, (*c*) the fuel-air ratio, pounds per pound, (*d*) the pounds of water vapor formed per pound of fuel, and (*e*) the dew-point temperature of the wet exhaust gases, if atmospheric pressure is 14.7 psia.

22. Same as problem 21 except that 20 per cent excess air by weight is used.

23. C_7H_{16} is burned with 20 per cent deficiency of air. Find (*a*) the combustion equation, assuming all H_2 is burned; (*b*) per cent CO_2 as read by an Orsat; (*c*) the air-fuel ratio, pounds per pound; (*d*) pounds of water vapor formed per pound of fuel burned; and (*e*) partial pressure of water vapor formed, psia, if atmospheric pressure is 14.7 psia. (*b*) 6.49 per cent, (*c*) 12.1 lb per lb, (*d*) 1.44 lb per lb, (*e*) 2.45 psia, *Ans.*

24. Fuel oil composed of $C_{16}H_{32}$ is burned with the chemically correct air-fuel ratio. Find (*a*) pounds of moisture formed per pound of fuel; (*b*) partial pressure of water vapor, psia; (*c*) CO_2 in stack gases as read by an Orsat, per cent; and (*d*) volume of stack gases, cubic feet per pound of oil, if temperature = 540 F and pressure = 14.7 psia.

25. The Orsat analysis of exhaust gas from an internal-combustion engine shows, percentage by volume, CO_2, 12.1; O_2, 0.3; CO, 3.5; H_2, 1.3; CH_4, 0.3; N_2, 82.5. Find (*a*) the probable combustion equation, (*b*) the probable fuel formula, and (*c*) the air-fuel ratio. (*c*) 13.3 lb per lb, *Ans.*

26. One pound of C_6H_{14} is burned with 80 per cent theoretical air at 14.7 psia. If only CO_2, CO, H_2O, and N_2 are formed, find (*a*) the combustion equation; (*b*) per cent by volume of CO in dry products; and (*c*) partial pressure of CO_2 in-products, assuming no condensation, psia.

27. Orsat data from an unknown fuel burned in a Diesel engine under light load are, percentage by volume: CO_2, 3.5; O_2, 16.0; CO, 0.0; N_2, 80.5. Determine (*a*) probable composition of the fuel; (*b*) air-fuel ratio, pounds per pound; and (*c*) per cent excess air.

CHAPTER 6

Steam Boilers and Generators

99. Development and Trends. A steam-generating unit is a combination of apparatus which may include any or all of the following: boiler, furnace, fuel-burning equipment, waterwalls, steam purifier, superheater, attemperator (a device for controlling steam temperature), steam reheater, economizer, and air heater.

Early steam boilers were limited to extremely low pressures (10 to 75 psig) because of their designs, materials of construction, and the facilities available for their fabrication. Industrial plants use steam at pressures usually less than 300 psig, as in many cases low-pressure steam is desirable for space heating and for the processing of materials. One boiler manufacturer lists boiler pressures ranging from 950 psig to 2075 psig among recent installations of its units in public-utility plants. The largest of the boilers mentioned operates to produce 1,200,000 lb of steam per hr when delivered at 2075 psig and 1000 F, and is capable of reheating steam to 1000 F. The lowest-pressure unit included produces 600,000 lb of steam per hr at 950 psig and 900 F total temperature. The general trend is to install one steam generator per turbine power unit. The greatest capacity for a single boiler unit under construction, at this time, is 1,400,000 lb per hr. Although one station is being built to operate with boiler-design pressure at 4500 psig, the present practice seems limited to 2350 psig pressure, and 1100 F total steam temperature with reheat to 1050 F. A unit is now under construction for 1200 F total steam temperature. In the higher steam pressure ranges the Mollier diagram of Fig. 39 indicates that an increase of 100 F steam temperature renders more energy available per pound of steam than is the case with only an increase of 100 psia in the pressure of the steam.

100. Classification. Steam boilers may be classified according to the relative position of the water and the hot gases as water-tube and fire-tube; arrangement of tubes, as vertical, horizontal, and inclined; curvature of tubes, straight-tube and bent-tube; and nature of service, stationary, portable, locomotive, and marine. The selection of a steam

boiler for a given service depends on the fuel available, type of service, capacity required, probable life of the installation, and other economic factors.

101. Fire-Tube Boilers. In the fire-tube boilers the hot gases pass through the tubes which are surrounded by water. Small vertical fire-

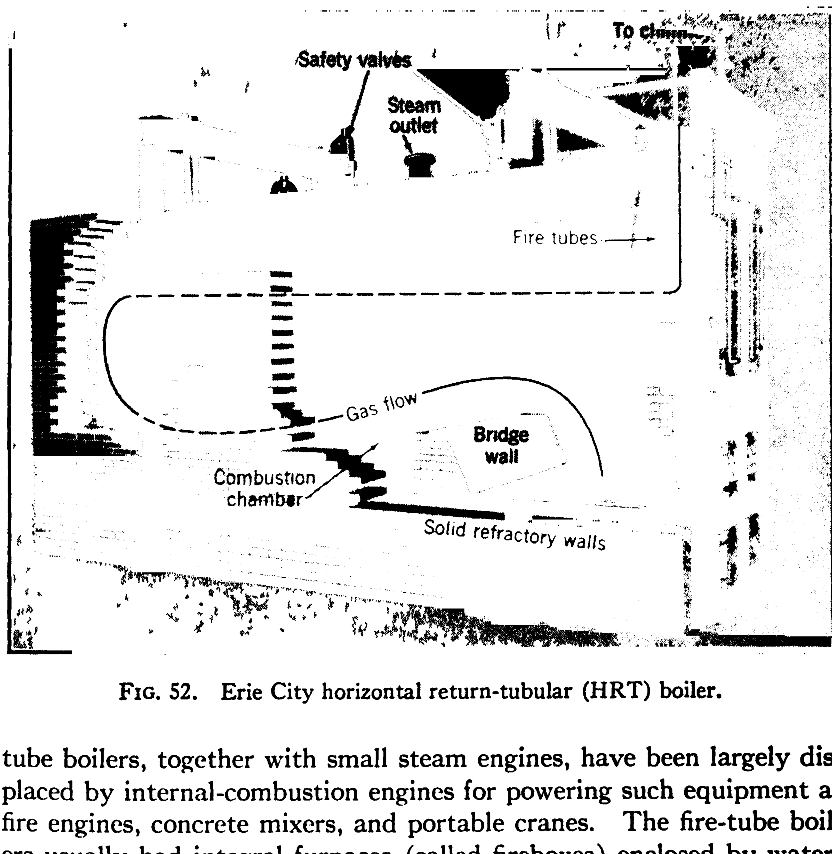


FIG. 52. Erie City horizontal return-tubular (HRT) boiler.

tube boilers, together with small steam engines, have been largely displaced by internal-combustion engines for powering such equipment as fire engines, concrete mixers, and portable cranes. The fire-tube boilers usually had integral furnaces (called fireboxes) enclosed by water-cooled surfaces. At present horizontal fire-tube boilers, with integral furnaces, are used for low-pressure heating or in somewhat larger sizes to provide relatively low-pressure steam for power and heating.

The stationary horizontal return-tubular (HRT) boiler, one type of fire-tube boiler, is shown in Fig. 52. It consists of the combination of grate, firebrick bridge wall, fire and ash doors, ashpit, and combustion space. The inside surfaces of the furnace walls are lined with firebrick. The boiler shell is supported by steel beams and hangers; lugs resting on the brickwork may be used for small (less than 60-in. diameter)

HRT boilers. The hot gases pass over the bridge wall, along the bottom of the boiler shell, and then return through the fire tubes to the front of the boiler. From the front end of the tubes the products of combustion pass through the breeching to the stack. The tube sizes may be 3, 3½, or 4 in. outside diameter, depending on the fuel to be used. Standard sizes of HRT boilers are built in accordance with the ASME (American Society of Mechanical Engineers) Boiler Code. Their shell diameters range from 36 to 84 in., and the corresponding tube lengths from 10 to 20 ft with steam pressures up to 175 psig and heating surfaces[1] ranging from 150 to 2500 sq ft. Return-tubular

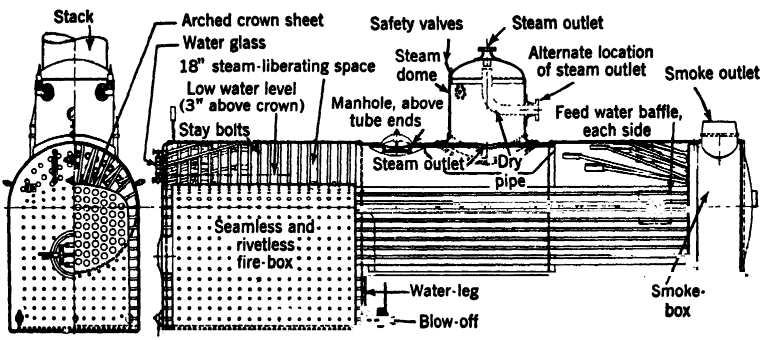


FIG. 53. Locomotive-type boiler, primarily for stationary service.

boilers are not suitable for large power plants because of their small capacities, limited steam pressures, and low rates of steaming capacity. They may be used in small industrial plants because of their low first cost, compactness, ease of cleaning, and water-storage capacity. They cannot be built for high pressures and capacities because the thickness of the outer-shell plates becomes excessive. Their limitations are much the same as those of all fire-tube boilers.

The portable locomotive-type of boiler, Fig. 53, has an internal furnace, surrounded on four sides by water-filled spaces (water legs). The top of the furnace, called the *crown sheet*, is below the water level. The gases make one pass from the furnace to the smoke outlet through straight fire tubes. Figure 53 indicates the stay bolts and braces necessary to hold in place flat and curved surfaces subjected to steam pressure and not supported by the fire tubes. Locomotive-type boilers do

[1] Boiler heating surface consists of that portion of the surface in contact with the water being heated on one side and the gas or refractory being cooled on the other; this surface is measured on the higher-temperature side. All stationary boilers are rated either on the basis of 10 sq ft of heating surface per boiler horsepower or on the weight of steam produced per hour under prescribed conditions of operation.

not operate with hot gases in contact with the outer shell; therefore, they may function with higher working pressures than those fire-tube boilers which have a portion of their outer shells exposed to the radiant heat of a burning fuel and the action of hot flue gases. The boiler of Fig. 53 may be used in oil-field operations with steam pressures up to

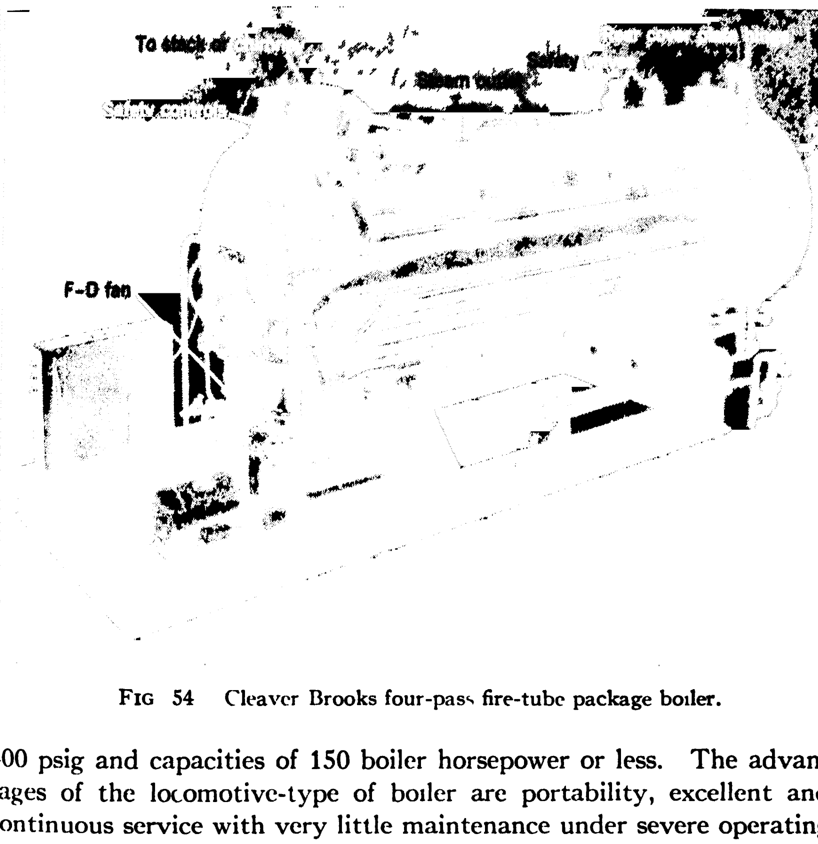


FIG 54 Cleaver Brooks four-pass fire-tube package boiler.

400 psig and capacities of 150 boiler horsepower or less. The advantages of the locomotive-type of boiler are portability, excellent and continuous service with very little maintenance under severe operating conditions, and large capacity for its size. The disadvantages are a slow rate of water circulation and large flat metal surfaces which require considerable stiffening with stay bolts and braces.

Figure 54 shows a modern package four-pass steam generator of the horizontal fire-tube type. This generator is designed for burning either oil or gas and is fully equipped with automatic controls, forced- and induced-draft fans, and other accessories. The unit is mounted on a base to aid in shipping and installing. The combustion chamber extends the length of the drum, and the gas flow is directed by division plates so that the gases make four passes through the drum.

The tubes in a steam generator of this type are accessible through removable front and rear cover plates or doors. Good mixing is obtained by providing swirl vanes for the entering air. The fire tubes serve to hold the tube sheet against internal pressure but stay bolts are necessary through the steam space near the top of the drum. Boilers

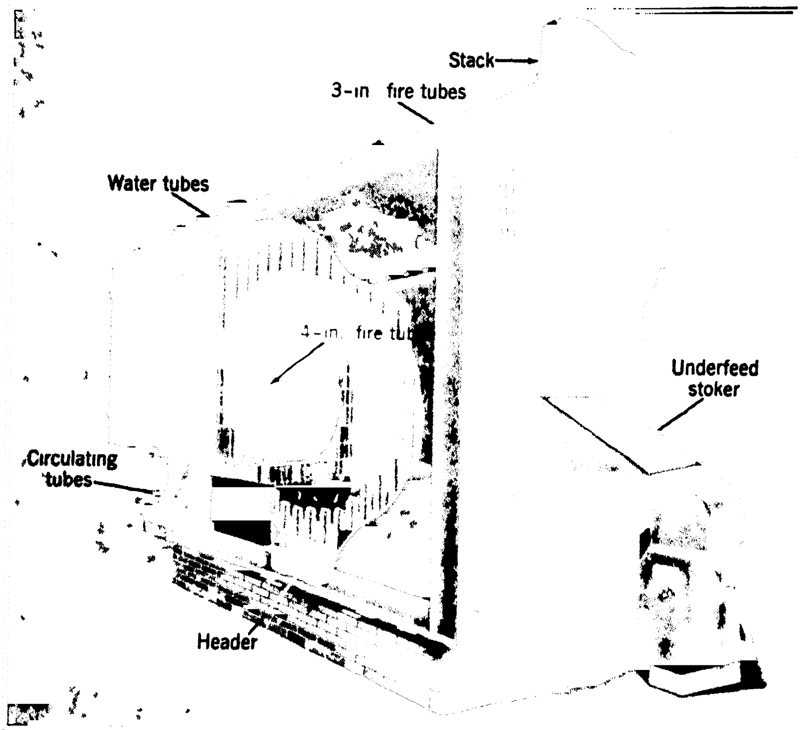


FIG. 55. Erie City fire- and water-tube boiler.

of this type are made in sizes ranging from 15 to 500 hp and 15 to 250 psig.

102. Water-Tube Boilers. In the water-tube construction the tubes carry water or steam, and the hot gases pass over their outer surfaces in contrast to the fire-tube arrangement described in Art. 101. Water-tube boilers are used almost exclusively when high pressures and high efficiencies are desired, since the stresses due to high pressure exist as hoop tension in the tubes rather than compressive stresses as in the fire-tube arrangement. The cleaning of water-tube boilers can be effectively done since the scale forming on the water side may be removed by a water or air-driven tube cleaner.

The objectives in any boiler design are low cost, simple tube shapes, compactness, accessibility, effective heat transfer, good circulation, and high capacity. This wide range of objectives has resulted in many designs and modifications of the water-tube boiler such as straight-tube, bent-tube, single-drum, multiple-drum, horizontal-drum, and cross-drum. Some of these are illustrated in Figs. 56–61.

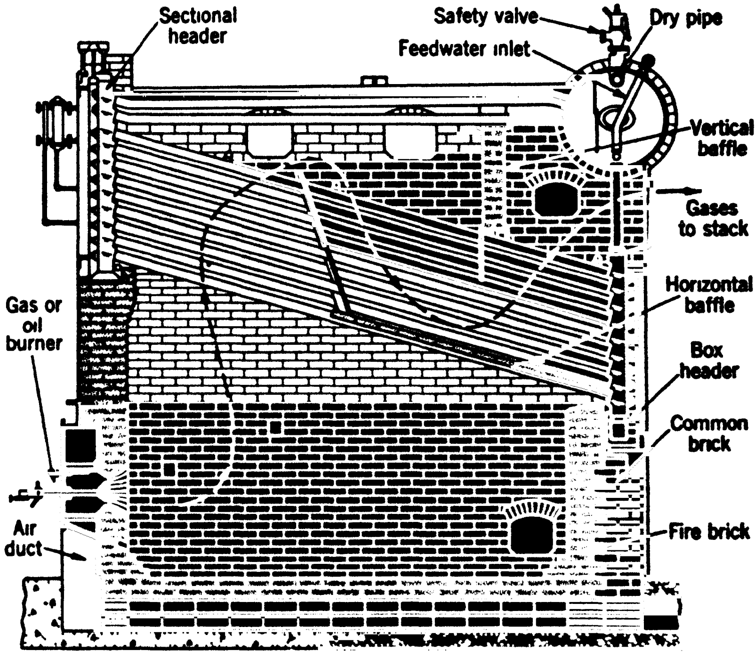


FIG. 56 B and W cross-drum straight-tube boiler

Figure 55 shows the design of a boiler incorporating both fire and water tubes. This type of boiler is adaptable to firing with oil or gas as well as coal. The ram-type underfeed stoker as shown represents a typical coal-firing arrangement. Front and rear cover doors allow accessibility for cleaning. The side-wall water tubes connect with horizontal headers, which in turn are connected with the water space in the drum. Good circulation is obtained as steam and water rise vertically in the tubes. Circulating tubes supply water from the rear of the boiler to the lower header. The vertical tubes absorb radiant heat and protect the refractory lining.

103. Horizontal Straight-Tube Boilers. A variation of the water-tube boiler employing straight tubes is shown in Fig. 56. The single

drum may be placed longitudinally or at right angles to the tube bank. The feedwater, introduced into the lower part of the drum, circulates downward through the rear tube headers, upward through the inclined tubes where steam bubbles form. The steam and water mixture rises rapidly through the front headers, thereby inducing circulation back to the drum, where separation of steam and water occurs.

The 4-in. water tubes are inclined horizontally at an angle of about 22 deg. The products of combustion are guided over the tube surfaces by baffles. The boiler is supported by a steel frame and is enclosed by solid brickwork. The working pressures of such units may be as high as 500 psig, and their heating surfaces may range from 750 to 15,000 sq ft. Superheaters are frequently used in this type of setting, and the unit is adaptable to power production.

Straight-tube boilers have an advantage in that all main tubes are alike and few special shapes are required. Certain disadvantages are present since the straight tubes must terminate in a header, the walls of which must be at right angles to the center line of the tube. This right-angle arrangement is necessary in order that a steamtight connection can be made by expanding the ends against the header wall. Access to the tubes for cleaning or replacement therefore requires a removable cover plate (handhole cover) at each end of the tubes. The handholes are usually elliptical so that the cover can be inserted through the opening and held mainly by the steam pressure from the inside.

104. Central-Station Steam Generators. Figure 57 shows a central-station steam generator designed to deliver 120,000 lb of steam per hr at 615 psig and 830 F total temperature at the superheater outlet. The unit is shown with a spreader stoker but is equally adaptable to oil, gas, or pulverized coal. The two-drum bent-tube steam boiler is provided with a forced-draft and an induced-draft fan, a tubular air preheater, centrifugal dust collectors, tubular-type economizer as well as a low-temperature and a high-temperature superheater of the convection type.

The upper drum is equipped with steam washers and driers through which the saturated steam must pass on entering the low-temperature superheater. The tubes of this superheater are individually attached to the drum; thus no inlet header is needed. As the steam passes from the low- to the high-temperature superheater header, condensate feedwater is injected in varying amounts (3000 lb per hr as a maximum) under boiler feed-pump pressure to control the final superheat temperature. This control maintains a temperature of 830 F plus or minus 2 deg in the steam leaving the high-temperature superheater even though the steam demand may vary from 50 to 100 per cent of rating. The high-temperature superheater is exposed to some radiant effect, thus

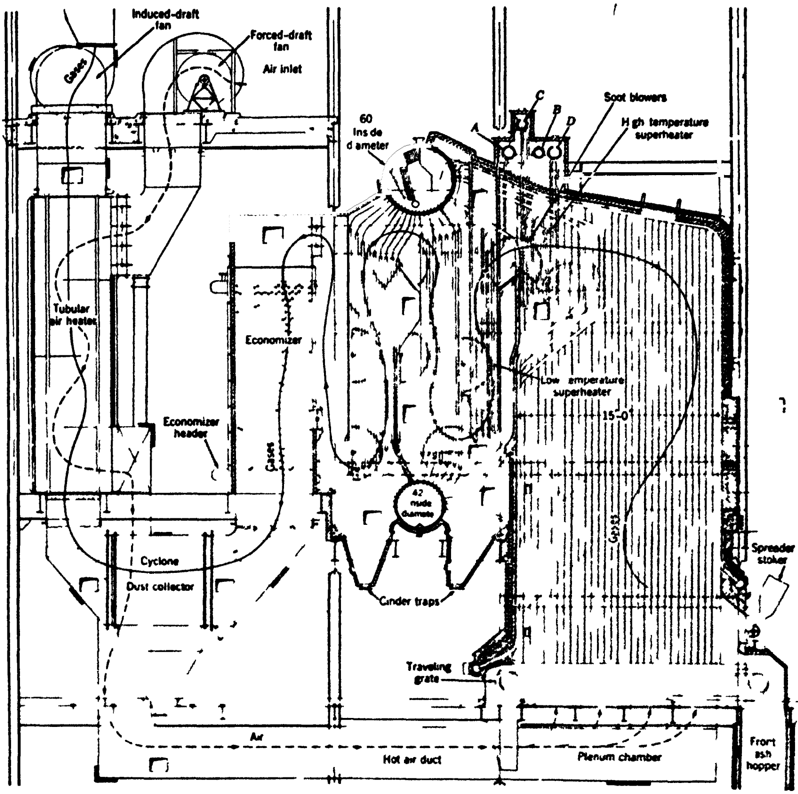


FIG. 57 Springfield two drum boiler

helping to give a more uniform superheat temperature with load variation

Figure 58 illustrates a single-drum central-station boiler The overall height of the generator may reach 150 ft. Steam capacities exceeding 1,350,000 lb per hr and pressures of 2300 psig and more are used with temperatures of the order of 1050 F Superheat and reheat are controllable Air entering the forced-draft fans is sent through the rotary air preheaters and is then carried to the front of the furnace where it is distributed to the coal burners as secondary air. The primary air is forced through the coal mills and carries the pulverized coal to the burners.

Figure 59 shows an open-pass boiler designed for a maximum capacity of 900,000 lb per hr with an operating pressure of 950 psig and a steam temperature of 950 F. In this unit the gases make three passes through the boiler as indicated by the arrows Since all the boiler tubes except

the middle row, which serves as a baffle, are placed in the side walls, resistance to gas flow is minimized, as well as tube cleaning. The boiler is applicable to slag-tap or molten-ash removal and does not require excessive headroom. The unit is fired with pulverized coal through horizontal cross-tube burners.

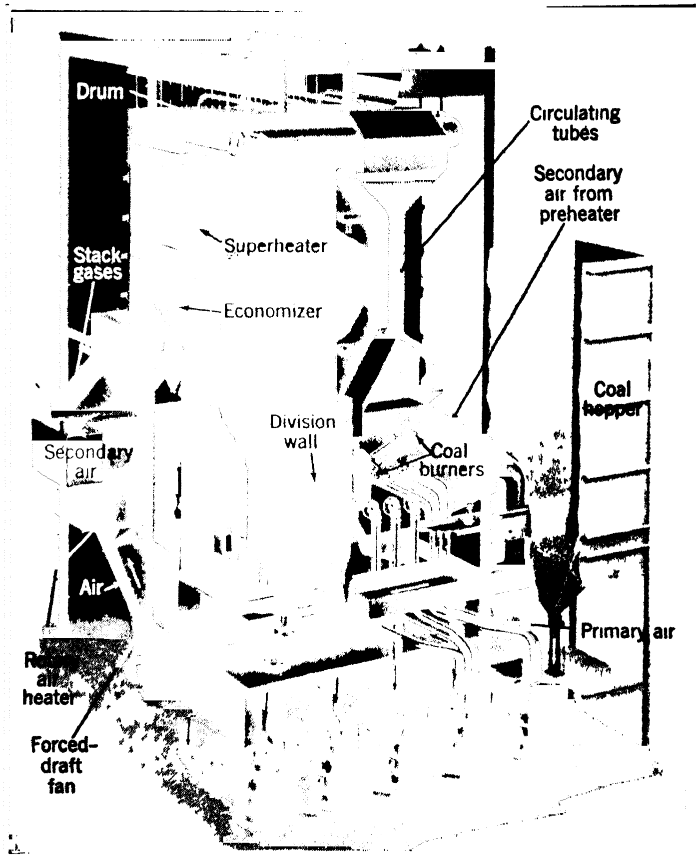


FIG. 58. B and W single-drum twin-type bent-tube radiant boiler.

105. Marine Steam Generators. Figure 60 illustrates a typical marine boiler designed for compactness and low headroom. Feedwater enters the economizer and then passes to the distribution pipe in the bottom of the upper drum. Circulating tubes carry water downward on the outside of the furnace to the lower drums, and as steam is generated in the main bank of tubes water rises to replace it. Saturated steam from the dry pipe is carried to the superheater inlet header at

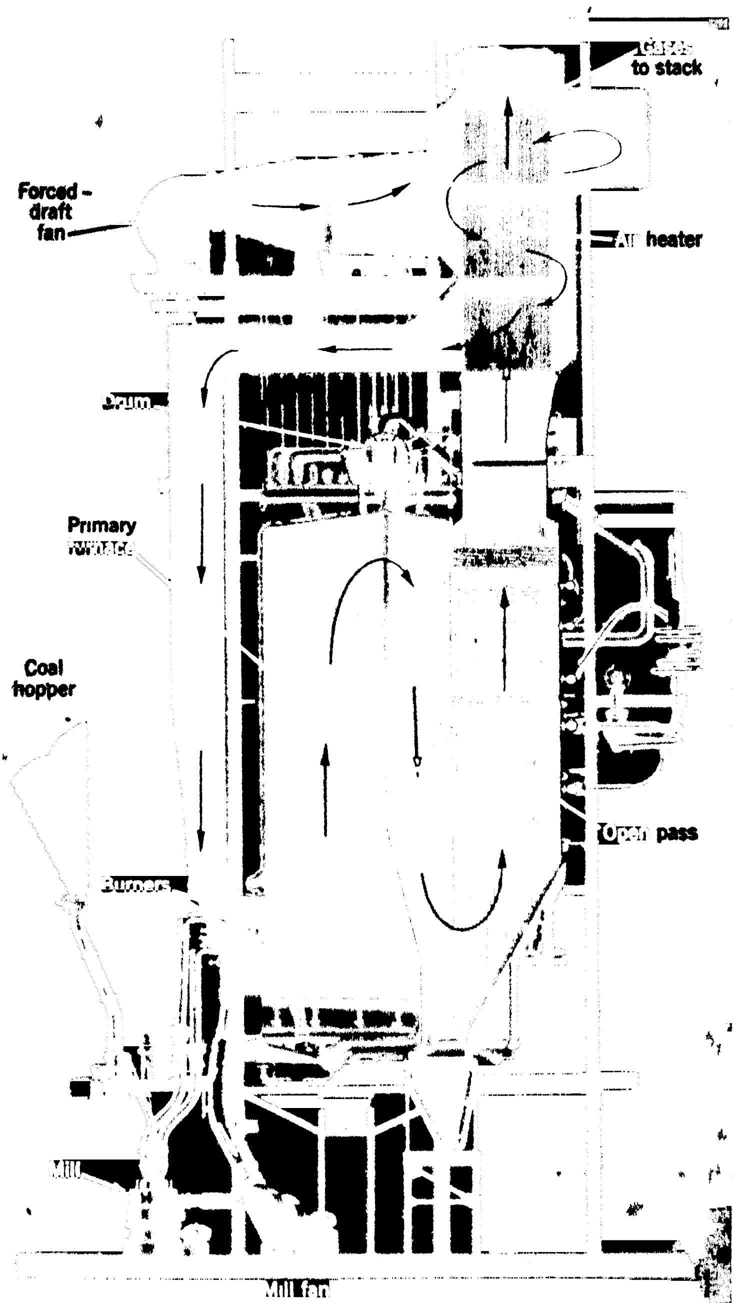


FIG. 59. B and W open-pass boiler.

the far end of the boiler. The degree of superheat may be accurately controlled by regulating the burners in the separately fired superheater. Saturated and superheated steam can be obtained simultaneously at any operating condition.

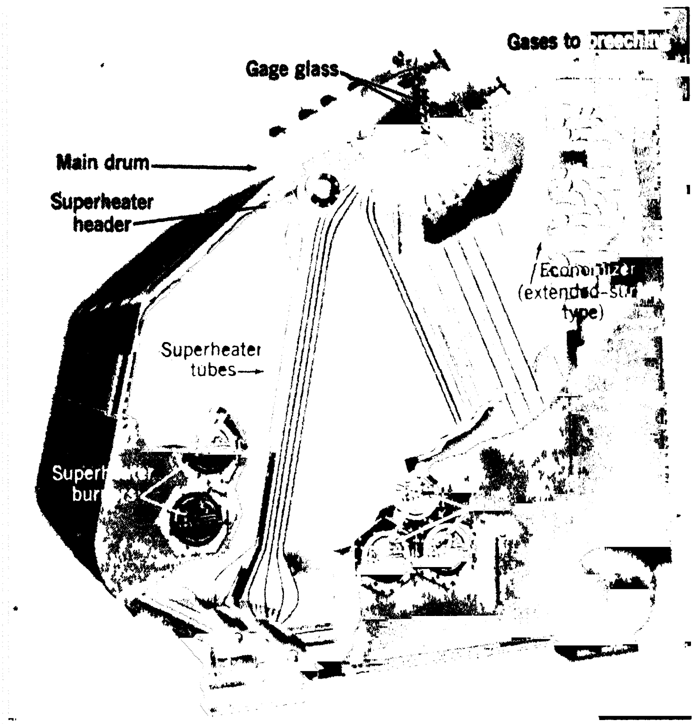


FIG. 60. Foster Wheeler marine steam generator, separately fired superheater type.

106. Forced-Circulation Steam Boilers. When forced circulation is introduced in a boiler in place of gravity circulation the tube diameter can be reduced, the tube circuit lengthened, and the wall thickness decreased for a given pressure. The tubes may be arranged as a continuous coil forming the furnace lining. Heat transfer is improved, a minimum of space is required, and headers and drums are no longer essential. In the Benson and Sulzer boilers, which employ forced circulation, the water makes one pass through the tubes and emerges as steam at the other end. Normally operating near 2000 psig, accurate and sensitive controls are necessary to regulate firing rate and water flow in order to obtain steam of the desired superheat.

Assisted-circulation boilers employ a pump to aid in circulation of

the water or the steam. The La Mont and the Velox boilers are of this type. In the Velox boiler, which has received wide acceptance in Europe, gases in the combustion chamber are pressurized to about 30 psig by a blower which is driven by a gas turbine operating from the

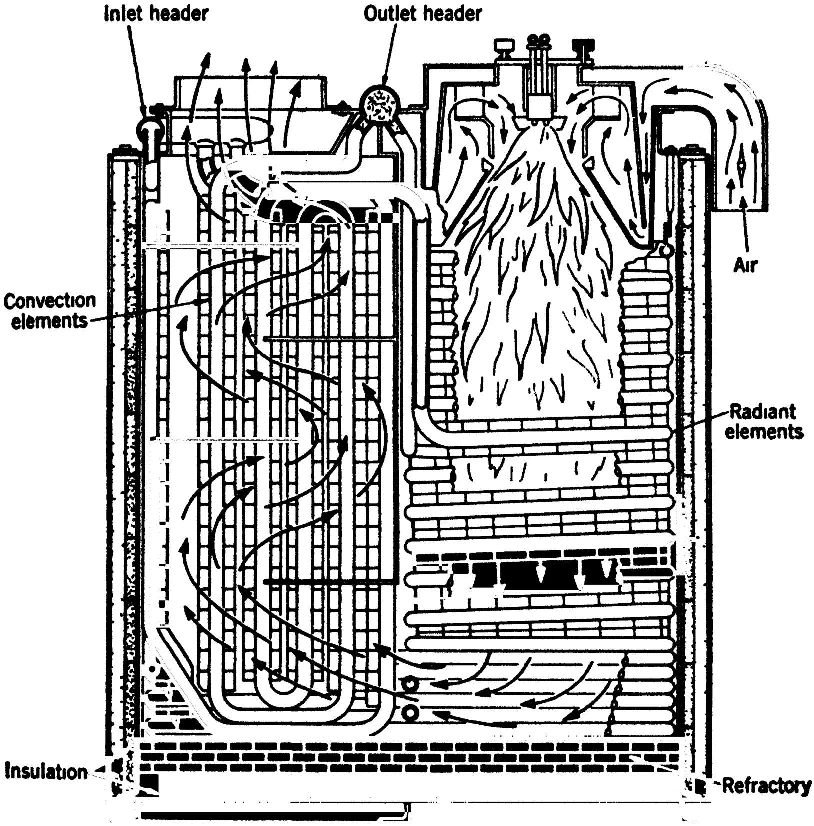


FIG. 61 C-E forced circulation boiler.

boiler exhaust gases. The improved heat transfer resulting from the increased density of the gases gives heat transfer rates that are reported as ten times those of conventional furnaces; thus the unit requires about one-fourth the space and weighs one-seventh as much as conventional units of the same capacity. Quick response to controls and rapid starting are other advantages claimed.

Figure 61 illustrates a forced-circulation boiler of the recirculation type. The discharged steam and unevaporated water enter a vertical drum in which centrifugal flow channels direct the water to the bottom

of the drum and the saturated steam rises from the center. The water re-enters the circulation pump to be forced again through the generator. A boiler of this type producing 6000 lb of steam per hr requires floor space approximately 7 ft square.

107. Waste-Heat Boilers. In many plants the exhaust gases from some process may be sufficient to supply the plant with steam. The

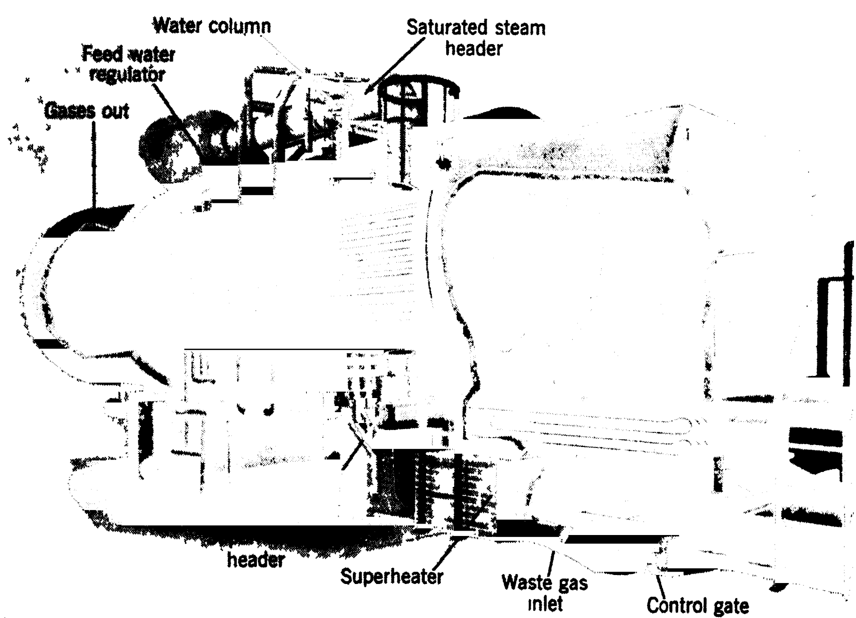


FIG. 62. Erie City waste-heat fire-tube boiler.

exhaust from internal-combustion engines ranges from 300 to 800 deg F, depending on the operating load and the type of engine (two- or four-stroke-cycle). Oil refinery stills may discharge gases continually at 900 to 1000 F, and gases from open-hearth steel furnaces may discharge at 1200 to 1300 F. The energy in these gases can be absorbed by a heat exchanger placed in the flow path. Both water-tube and fire-tube designs are used. Figure 62 shows a waste-heat boiler (fire-tube) receiving gases from ducts running beneath the floor. A superheater is located in the gas stream directly above the control gate.

Waste-heat boilers must rely on convection rather than radiation as a means of heat transfer since the surfaces are not exposed to the flame. Small tubes (1½ to 2 in.) in compact arrangement and high gas veloci-

ties across the tubes are desirable; however, resistance and draft loss are increased. Induced-draft fans can be used to overcome draft losses and maintain high velocity through the tubes. Approximately 60 per cent of the energy in the gases measured between the temperature at the inlet and that of the saturated steam is available; this available energy divided by the Btu that must be added to each pound of feed-water will give the pounds of steam produced by the waste-heat boiler.

108. Trends in Steam Generator Design. In large central stations it is not uncommon for the plant to consume well over a trainload of coal per day. The economics of such an installation shows that major expenditures are justified if the efficiency can be improved by a very small amount. This fact has been responsible for a trend in central-station generators leading to high pressures, high temperatures, and the application of superheaters, reheaters, pulverized coal firing, automatic combustion controls, and other features by which the efficiency of the steam cycle may be improved.

Drums are expensive components for high-pressure boilers. Since the bursting stress in a drum is directly proportional to the radius and the operating pressure, the wall thickness necessary after tube openings have been made may reach 6 in. for high-pressure drums. Fabrication of drums by welding requires that in the interest of safety the joints be X-rayed for detecting possible flaws. The number and size of drums is therefore kept to the minimum. Drums are used in conventional boilers to provide separation space between the steam and the water and as a terminal for tubes, thereby eliminating headers and handholes. Access doors or manholes in the drum ends allow entrance for assembly, inspection, and maintenance. Tube holes are drilled radially in the drum; therefore practically all tubes terminating at the drum require a radius bend. The number and arrangement of drums for a given installation depend on such factors as (1) efficiency, (2) type of fuel, (3) nature of load, (4) operating pressure, (5) future requirements, (6) headroom available, and (7) climatic conditions.

In the interest of efficiency adequate tube area must be exposed to the combustion products. If smaller tubes are used, a larger surface area in a given space can be obtained and tube thickness can be less for a given pressure; however, if natural circulation is to be obtained, tube runs must be shorter; consequently, more headers or drums are required. Closely spaced tubes giving high turbulence provide good heat transfer, but they increase resistance to gas flow, and more fan power is required.

To lower construction costs outdoor power plants are being built with sheltered space only for the operators and such equipment as requires

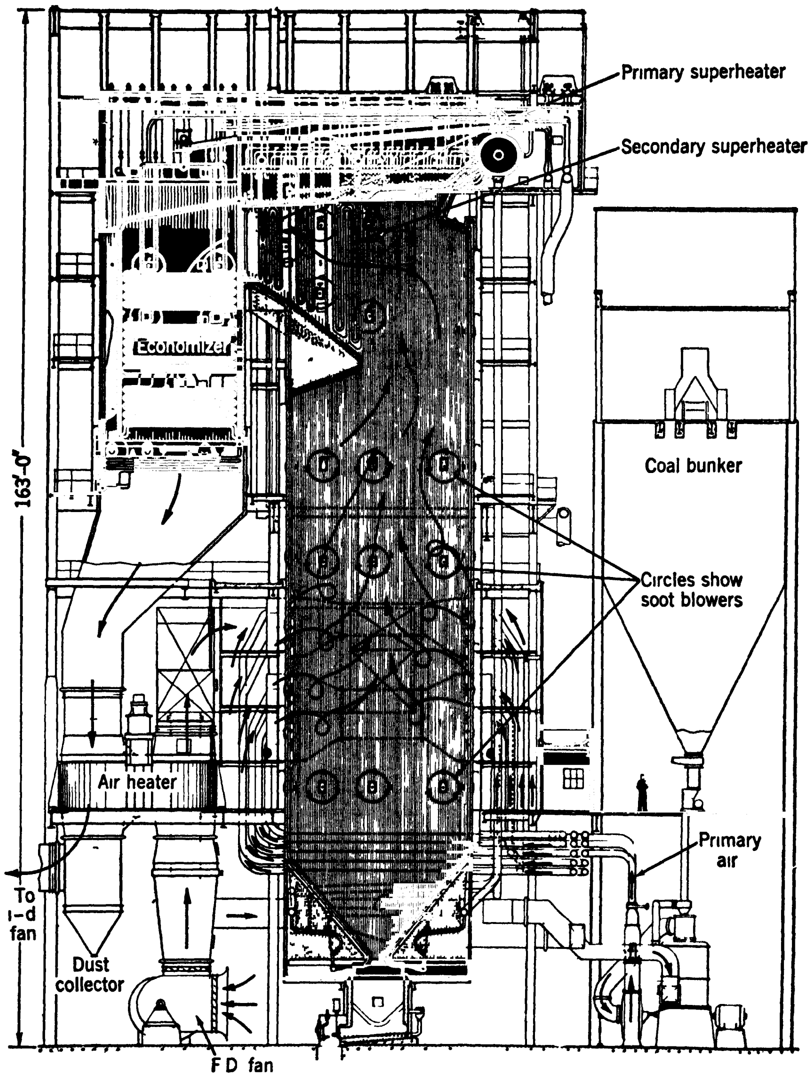


FIG 63 C-E radiant boiler.

protection from moisture. Such installations allow high drum settings in which good natural circulation can be produced, since the circulation flow is proportional to the height and the difference in density between the water in the circulating tube and the steam in the boiler tube. Low resistance to gas flow can be obtained by radiant heat transfer rather than by convection. Therefore tall open combustion spaces are used

in which most of the tubes are placed in the side walls rather than across the gas stream.

An example of central-station practice is found in the Joppa Steam Station of Electric Energy, Inc. The plant consists of four independent units, each one comprised of a steam generator, a turbine generator set, and a condenser plus the usual auxiliaries. Each of the four identical units (not interconnected except for minor auxiliaries) is designed to produce 163,000 kw.

The steam generators, Fig. 63, stand 163 ft in over-all height. They are single-drum radiant-type integral-furnace boilers. Each is designed to deliver 1,250,000 lb of steam per hr at 1925 psig when fired with coal of 10,000 Btu per lb, 15 per cent moisture, and 15 per cent ash. Steam is delivered at 1055 F and reheated to 1005 F after partial expansion in the turbine. Five Raymond bowl-type mills, Fig. 78, serve each boiler although one is a spare.

109. Boiler Capacity. The output of a steam-generating unit is frequently given as the number of pounds of steam delivered per hour. Since steam at various pressures and temperatures contains varying amounts of energy, the number of pounds of steam generated is not an exact measure of the energy produced. The capacity of a steam boiler is more definitely expressed as the total heat transferred by the heating surfaces in Btu per hour. As this figure is numerically large, the ASME recommends a *unit of heat absorption* of 1000 Btu called the kilo Btu (kB) per hour or a unit of 1,000,000 Btu called the mega Btu (mB) per hour. The heat-transfer process that occurs in a steam generator is a steady-flow process for which the transferred heat is equal to the change in the enthalpy of the fluid. Therefore the boiler output, as measured by the heat absorbed by water and steam, is

$$Q = \frac{m_s(h - h_f)}{1000} \quad \text{kB per hr} \tag{171}$$

where Q = boiler output, kB per hr.

m_s = weight of steam delivered by boiler (or superheater, if used), lb per hr.

h = enthalpy of 1 lb of steam at observed pressure and quality or temperature, Btu per lb.

h_f = enthalpy of the liquid of 1 lb of feedwater at observed conditions as water reaches the boiler (or economizer, if used), Btu per lb.

When the weight of steam m_s is the maximum amount that the boiler can deliver per hour at the specified operating temperature, equation

171 will give the *maximum capacity*. When m_s is the weight of steam that the boiler can deliver most efficiently, the corresponding output may be called the *normal capacity*.

The *boiler horsepower* as originally standardized by the ASME in 1889 was based on the requirements of a steam engine that used 30 lb of steam per horsepower-hour at 70 psig and boiler feedwater at 100 F. This corresponds to the evaporation of 34.5 lb of water per hour "from and at" 212 F (33,475.35 Btu per hr), which means the conversion of *34.5 lb of water per hour from water at 212 F into dry steam at 212 F* and, therefore, at normal atmospheric pressure (14.7 psia). Each pound of steam generated under these conditions requires the enthalpy of evaporation at normal atmospheric pressure, which is 970.3 Btu. Hence the capacity of a boiler may be expressed as

$$\text{Boiler horsepower} = \frac{m_s(h - h_f)}{970.3 \times 34.5} \tag{172}$$

where m_s, h, and h_f are as defined for equation 171. Although the term boiler horsepower is no longer considered an acceptable unit of boiler capacity, the general use of this term for many years requires that the engineer know its meaning and limitations.

Other units of boiler capacity that have been used for purposes of comparison are the factor of evaporation and the equivalent evaporation. The *factor of evaporation* is the ratio of the heat absorbed by 1 lb of feedwater under observed boiler conditions to that absorbed by 1 lb of water evaporated from and at 212 F. Hence

$$\text{Factor of evaporation} = \frac{h - h_f}{970.3} \tag{173}$$

The *equivalent evaporation* is defined as the pounds of water per hour which would have been evaporated from and at 212 F if the same amount of energy had been absorbed as under observed boiler conditions. Therefore

$$\text{Equivalent evaporation} = \frac{m_s(h - h_f)}{970.3} \tag{174}$$

A term used especially for small boilers is the *rated horsepower*. All small boilers are rated on a basis of 10 sq ft of heating surface per boiler horsepower. By this method, boilers of equal heating surface are given the same rating, but one boiler may have its heating surface arranged more advantageously than another and thus have a greater capacity under like operating conditions. The rated horsepower does not ex-

press the limitations of capacity of present-day boilers, as most power-plant boilers can develop from 400 to 600 per cent of their normal rating and some operate for long periods at 300 per cent rating. The efficiency of the boiler may be decreased when it develops much more than its rated horsepower; see Fig. 64. This decrease in efficiency is

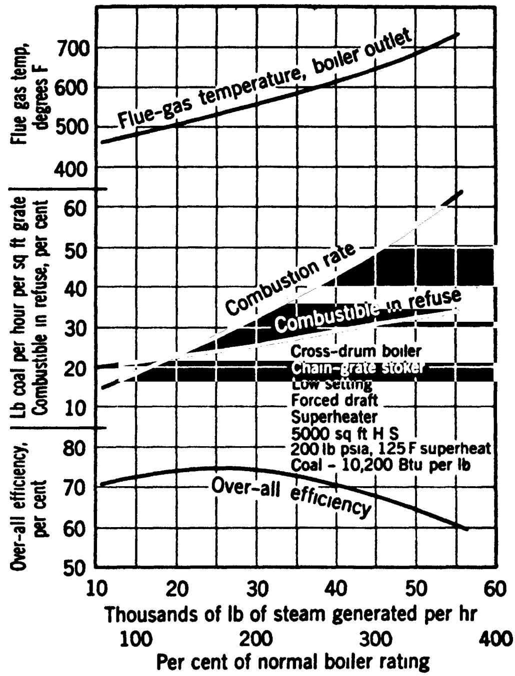


FIG. 64. Typical boiler performance curves, chain-grate stoker furnace.

generally not appreciable until the boiler is operating at 200 per cent or more of normal boiler rating, the decrease being due to the inability of the boiler to absorb the additional heat liberated by the fire at the higher ratings.

110. Boiler Performance. The performance of a steam-generating unit may be expressed in terms of pounds of steam generated, rate of combustion, heat transfer in Btu per square foot of heating surface per hour, temperature of flue gases, percentage of CO_2 in the flue gases, unburned combustible in the ash, percentage of boiler rating developed, and over-all efficiency.

The over-all *boiler efficiency* of a steam-generating unit at any operating condition is the ratio of the transferred heat to the energy sup-

plied as fuel, i.e., output divided by input, or

$$e_b = \frac{m_s(h - h_f)}{m_f \times F} \times 100 \tag{175}$$

where e_b = efficiency of steam-generating unit, per cent; includes boiler, superheater, furnace, waterwalls, air heater, and economizer.

m_f = total weight of fuel fired per hour, lb; cu ft per hr for gas.

F = high heat value of fuel as fired, Btu per lb; Btu per cu ft for gas.

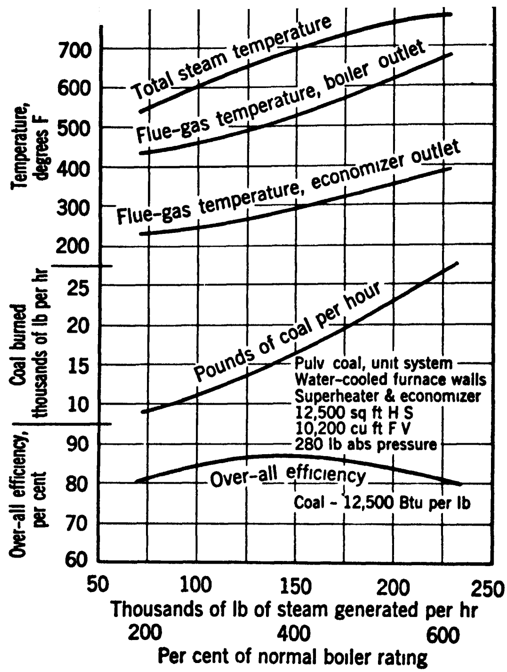


FIG. 65. Typical boiler performance curves, pulverized-coal furnace.

The rate of combustion may be expressed in pounds of fuel burned (*a*) per square foot of projected grate surface per hour, or (*b*) per cubic foot of furnace volume per hour. The boiler capacity (pounds of steam generated per hour) that can be developed is dependent on the rate of combustion, i.e., on the kind of fuel, size and kind of grate, and quantity of air supplied. Rates of combustion as high as 200 lb of coal per sq ft of grate surface per hr have been obtained in locomotive-type boilers with a resultant high capacity but a lowered efficiency. In

modern steam power plants, the most desirable rate of combustion for a given boiler setting for maximum efficiency is determined experimentally. The efficiency increases with the boiler capacity and rate of combustion until the maximum is reached; beyond this point the efficiency decreases with increase in capacity; see Figs. 64 and 65. In Fig. 64 the efficiency peak occurs when the boiler is generating between 20,000 and 30,000 lb of steam per hr, beyond which the efficiency decreases because of the heat lost in increased flue-gas temperature and an increase in the combustible (unburned carbon) in the refuse. A larger boiler having a pulverized-coal furnace and an economizer showed an efficiency peak, Fig. 65, when producing 115,000 to 150,000 lb of steam per hr, which corresponds to a boiler rating of 300 to 400 per cent. Comparison of the curves of the flue-gas temperatures shows how the use of an economizer decreases the heat carried away by the flue gases.

Stoker-fired boilers may show from 60 to 85 per cent efficiency, depending primarily on the size of the unit, furnace design, and the number of heat-saving auxiliaries employed. Large pulverized-coal installations have shown boiler efficiencies of 80 to 88 per cent. If air preheaters and economizers are in use in such plants, over-all efficiencies of 85 to 92 per cent may be obtained at times.

Example. A steam generator evaporated 40,000 lb of water per hr from a feed-water temperature of 220 F to steam at 180 psia and a quality of 0.97. Weight of coal fired per hour, 4500 lb; high heat value of the coal as fired, 11,800 Btu per lb. Determine the rate of heat absorption in kilo Btu per hour, the boiler horsepower developed, and the efficiency.

Solution. The desired results are obtained by use of equations 171, 172, and 175.

$$Q = \frac{40{,}000[346.03 + (0.97 \times 850.8) - 188.13]}{1000}$$

$$Q = 39{,}327 \text{ kB per hr}$$

$$\text{Boiler horsepower} = \frac{40{,}000[346.03 + (0.97 \times 850.8) - 188.13]}{970.3 \times 34.5}$$

$$\text{Boiler horsepower} = 1174.8$$

$$e_b = \frac{40{,}000[346\ 03 + (0.97 \times 850.8) - 188.13]}{4500 \times 11{,}800} \times 100$$

$$e_b = 74.06 \text{ per cent}$$

111. Heat Balance of Steam-Generating Unit. The distribution of heat resulting from the combustion of fuel in a boiler furnace can best be studied by means of a heat balance, which consists of a tabulation of the heat absorbed by the steam-generating unit and of the various heat losses incident to the combustion of the fuel. Calculations are

generally made in terms of Btu per pound of fuel and in percentage; hence the totals should equal the high heat value of the fuel as fired and 100 per cent, respectively. The items considered may be listed as follows:

1. *Heat absorbed by steam-generating unit.* The heat absorbed by the steam-generating unit (including economizer, superheater, and reheater, if used) may be calculated by the following equation:

$$H_1 = \frac{m_s}{m_f}(h - h_f) \tag{176}$$

where H_1 = Btu absorbed by water and steam per pound of fuel, as fired. Other items are as for equations 171 and 175.

2. *Heat loss due to moisture in fuel.* The moisture in the fuel is evaporated and leaves the boiler setting in the form of superheated steam. The partial pressure of the superheated steam in the flue gases is assumed to be 1 lb per sq in. abs. The temperature is that of the stack gases.

$$H_2 = m_m(h'' - h_f') \tag{177}$$

where H_2 = Btu loss per pound of fuel, as fired.
m_m = weight of free moisture per pound of fuel, as fired.
h'' = enthalpy of superheated steam at flue-gas temperature and 1 psia, Btu per lb.
h_f' = enthalpy of liquid at temperature of fuel entering furnace, Btu per lb.

3. *Heat loss due to water from combustion of hydrogen.* The hydrogen in the fuel burns to water, which leaves the boiler setting in the form of superheated steam.

$$H_3 = 9H_y(h'' - h_f') \tag{178}$$

where H_3 = Btu loss per pound of fuel, as fired.
H_y = weight of hydrogen per pound of fuel, as fired.

4. *Heat loss due to moisture in air supplied.* This heat loss is small and may be computed thus:

$$H_4 = m_v \times 0.46(t_g - t_a) \tag{179}$$

where H_4 = Btu loss per pound of fuel, as fired.
m_v = percentage of saturation expressed decimally, times weight of water vapor required to saturate 1 lb of dry air (from Table XVIII, Appendix) at t_a, times weight of dry air used per pound of fuel, as fired.
0.46 = mean specific heat of steam from t_g to t_a.

t_g = flue-gas temperature at boiler outlet, deg F.
t_a = temperature of air entering furnace, deg F.

5. *Heat loss due to dry chimney gases.* The heat lost by the dry chimney gases is usually the largest loss and may be calculated by

$$H_5 = m_{dg}c_p(t_g - t_a) \tag{180}$$

where H_5 = Btu loss per pound of fuel, as fired.
m_{dg} = weight of dry gas at boiler outlet, pounds per pound of fuel, as fired; see equation 169.
c_p = mean specific heat of dry gas, about 0.24.

6. *Heat loss due to unburned gaseous combustible.* This heat loss is generally small and is caused by an insufficient air supply, some of the carbon in the fuel being burned to carbon monoxide.

$$H_6 = \frac{CO}{CO_2 + CO} \times 10{,}160 \times C_1 \tag{181}$$

where H_6 = Btu loss per pound of fuel, as fired.
CO and CO_2 = percentages by volume, determined by flue-gas analysis.
C_1 = weight of carbon actually burned per pound of fuel; see equation 163.

7. *Heat loss due to unconsumed combustible in refuse.* Some of the carbon in the fuel, either unburned or partly burned, may fall into the ashpit. This loss depends on the type of grate, rate of combustion, and size and kind of coal.

$$H_7 = \frac{14{,}540 m_r C_r}{m_f} \tag{182}$$

where H_7 = Btu loss per pound of fuel, as fired.
m_r = weight of refuse, lb per hr.
C_r = weight of carbon, lb per lb of refuse.

8. *Heat loss due to unconsumed hydrogen and hydrocarbons, radiation, and unaccounted-for losses.* These losses are determined by subtracting the heat absorbed by the boiler and heat losses 2 to 7, inclusive, from the high heat value per pound of fuel as fired. Unaccounted-for heat losses are those which cannot be measured.

$$H_8 = F - (H_1 + H_2 + H_3 + H_4 + H_5 + H_6 + H_7) \tag{183}$$

Example. The following data were obtained from a boiler test. Heat absorbed by boiler, 74.8 per cent of the high heat value of the fuel as fired. Analysis of coal as fired, percentage by weight: C, 62; H, 4; N_2, 1; O_2, 8; S, 2; moisture, 8; ash, 15;

and Btu per lb, 11,800. Flue-gas analysis, percentage by volume: CO_2, 13; CO, 1; O_2, 5; and N_2, 81.

Temperature of air and fuel entering furnace, 74 F; temperature of flue gas, 452 F; temperature of steam in boiler, 360 F; percentage of saturation of air entering furnace, 70 per cent; coal fired per hour, 2000 lb; refuse per hour, 420 lb; and unburned carbon in refuse, 18 per cent. Barometer, 29.92 in. mercury. Calculate and tabulate a complete boiler heat balance per pound of coal as fired.

Solution. Weight of carbon appearing in the flue gas, equation 163,

$$C_1 = \frac{(2000 \times 62) - (420 \times 18)}{2000 \times 100} = 0.5822 \text{ lb per lb of coal}$$

Weight of dry chimney gases, equation 169,

$$m_{dg} = \left[\frac{(4 \times 13) + 5 + 700}{3(13 + 1)}\right] \times 0.5822 = 10.494 \text{ lb per lb of coal}$$

Weight of dry air supplied, equation 170,

$$m_{aa} = 10.494 - 0.5822 + 8\left(0.04 - \frac{0.08}{8}\right) = 10.15 \text{ lb per lb of coal}$$

Boiler Heat Balance per Pound of Fuel As Fired

Item	Method of Calculation	Btu	Percentage
Heat absorbed by boiler	0.748(11,800)	8,826	74.80
Moisture in fuel	0.08(1265.9 − 42.03)	98	0.83
Hydrogen in fuel	9(0.04)(1265.9 − 42.03)	440	3.73
Moisture in air	0.70(0.01815)(10.15)(0.46)(452 − 74)	23	0.18
Dry chimney gases	10.494(0.24)(452 − 74)	952	8.07
Incomplete combustion	[1 ÷ (13 + 1)]10,160 × 0.5822	423	3.59
Combustible in refuse	(14,540)(420)(0.18) ÷ 2000	550	4.66
Radiation and unaccounted-for losses	By difference	488	4.14
	Total	11,800	100.00

The heat balance as ordinarily calculated gives only the distribution of the *actual losses* for operation of the steam generator over a limited period of time. It does not represent average operating conditions, since the various *stand-by losses* are not considered. These include (1) heat lost in shutting down boilers, (2) fuel required to start up cold boilers, (3) fuel burned in banking fires, and (4) heat discharged to water in "blowing off" and for cleaning boiler surfaces. Some of the actual heat losses may be considerably reduced or entirely eliminated; other losses are inherent and cannot be prevented. *Inherent losses* are the losses caused by the discharge of products of combustion (flue gases, moisture in fuel, moisture in air supplied for combustion, moisture from

the combustion of hydrogen) at a higher temperature than that at which they entered the furnace.

112. Boiler Accessories. The accessories usually provided for a boiler are steam-pressure gage, Art. 8, water column, feedwater regulator, safety valve, fusible plugs, blowoffs, soot blowers, draft gages, and boiler control system.

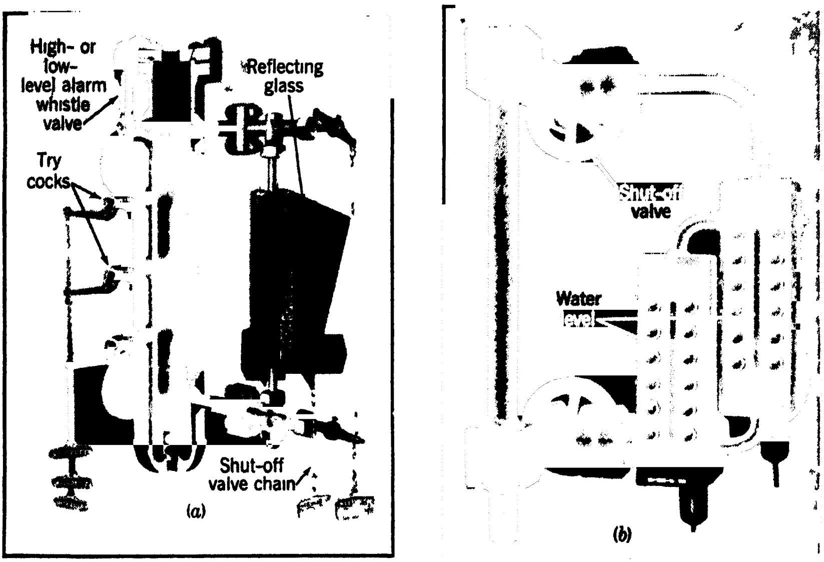


FIG. 66. (*a*) Reliance water column showing high- and low-level alarm. (*b*) Yarway high-pressure flat glass water gage

A water column of the type shown in Fig 66 is connected to the front of the boiler drum so that the water level may be visible from the boiler-room floor. The top of the water column is connected with the steam space, and the bottom to the water space in the boiler drum. The water column shown in Fig. 66(*a*) is provided with high- and low-water alarms. The action of the water column depends on the displacement of floats suspended from two levers, the whole assembly being in static equilibrium at normal boiler water levels with the alarm valve seated. Whenever high- or low-water conditions prevail, the equilibrium of the floats is changed; either condition opens the valve, and the released steam operates an alarm. Figure 66(*b*) is available for pressures as high as 2500 psig. The gage is illuminated so that the meniscus at water level is easily seen. The water level in most horizontal water-tube boilers is maintained at a point at which the drum

is about half-full of water. The water column is usually connected so that the gage glass is half-filled at the desired water level.

The purpose of a *feedwater regulator* is to supply, automatically, the same amount of water to a boiler as is being converted into steam, thereby maintaining a nearly constant water level. Several types of feedwater regulators are in common use: float-lever, thermoexpansion, thermohydraulic, and pneumatic. They are generally used in connec-

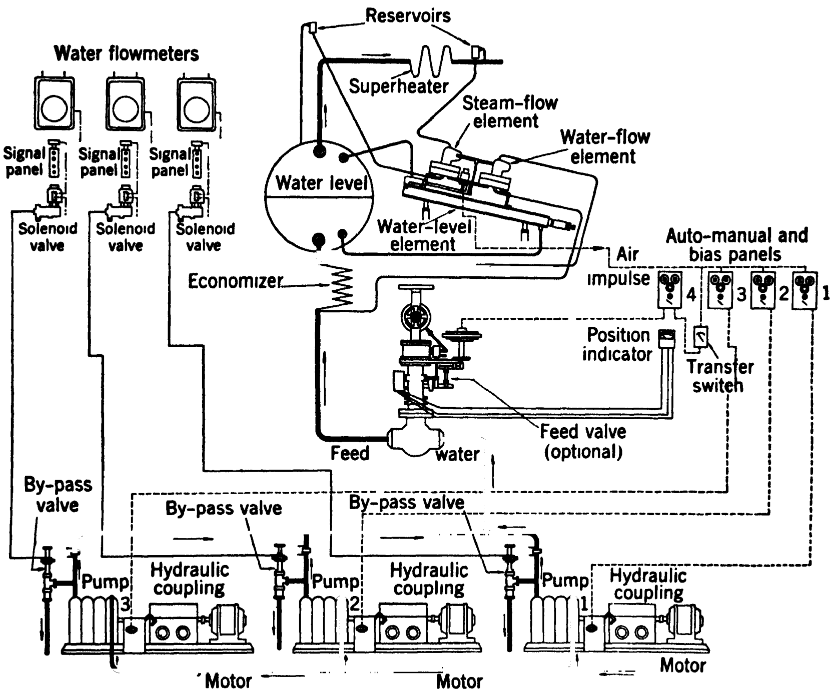


FIG. 67. Copes balanced-flow feedwater control.

tion with suitable pump governors. In the float-lever type the actuating member is a hollow float which rises and falls with the water level. When the water level falls the movement of the float opens the feedwater-control valve through levers. This type of regulator responds quickly to fluctuations of water level and is built for boiler pressures up to 700 psig.

The balanced-flow control, Fig. 67,[2] is responsive to three influences —rate of steam flow from the boiler, rate of water flow to the boiler, and drum water level. Steam flow is measured by variations in the

[2] Courtesy of Copes-Vulcan Division, Continental Foundry and Machine Company.

pressure drop between the boiler drum and the superheater outlet. Water flow is measured by variations in the pressure drop between the economizer inlet and the boiler drum. The two flow-measuring elements are mounted integrally on the frame of the thermostat.

The resultant of the two flow influences is balanced against that of the water-level influence, and the final resultant actuates an air pilot valve. Air impulses from this pilot valve pass through auto-manual panels 1, 2, and 3 to the individual controls on the hydraulic couplings of the respective boiler feed pumps. This operation parallels all three pumps, although only two are normally in service. Any pump may be put in or taken out of service at any time by operating its push-button control. The coupling controls can be set for the desired pump discharge pressure. The feed valve may also be operated by a handwheel. A minimum flow control is provided to protect the pumps against overheating on light loads.

Safety valves are used to protect boilers against excessive steam pressures, by opening automatically at a predetermined pressure and allowing the steam to escape. Figure 68 shows a section through a pop safety valve (only type sanctioned by ASME Boiler Code) in which the boiler pressure is resisted by a spring. The spring tension may be adjusted at the top, and a lever is provided for testing the valve manually. The number and size of safety valves for a given boiler are usually specified by city or state legislation or by insurance companies.

Boilers (especially fire-tube types) operating at less than 225 psig are generally protected by *fusible plugs*. They consist of steel or bronze bushings, filled with a tin alloy that melts at or about 450 F, and inserted in the boiler drum at the lowest permissible water-level location as listed by the ASME Boiler Code. The melting point of the tin is above the temperature of the steam and below the temperature of the hot gases. The small end of the plug is exposed to the products of combustion; when the water level is low enough to uncover the plug, the alloy melts and steam escapes. In this way, probable excessive boiler pressure is prevented, and the escaping steam attracts the attention of the attendant so that precautions may be taken to prevent overheating of the boiler metal.

Blowoff connections, as shown on the boiler in Fig. 53, attached to the lowest part of the boiler and sometimes also to the steam drum, are used for the removal of some water to carry off mud, sediment, scale, and scum. Impurities from the large quantities of water evaporated are constantly being precipitated. A continuous (bottom) blowoff with a small pipe is sometimes used to remove the impurities as they are precipitated. However, when this is done the larger blowoff

valves must be opened occasionally for a more thorough removal of the accumulated mud.

Feedwater impurities combine mechanically and chemically, during boiler operation, to form a layer of scale on the heating surfaces. The

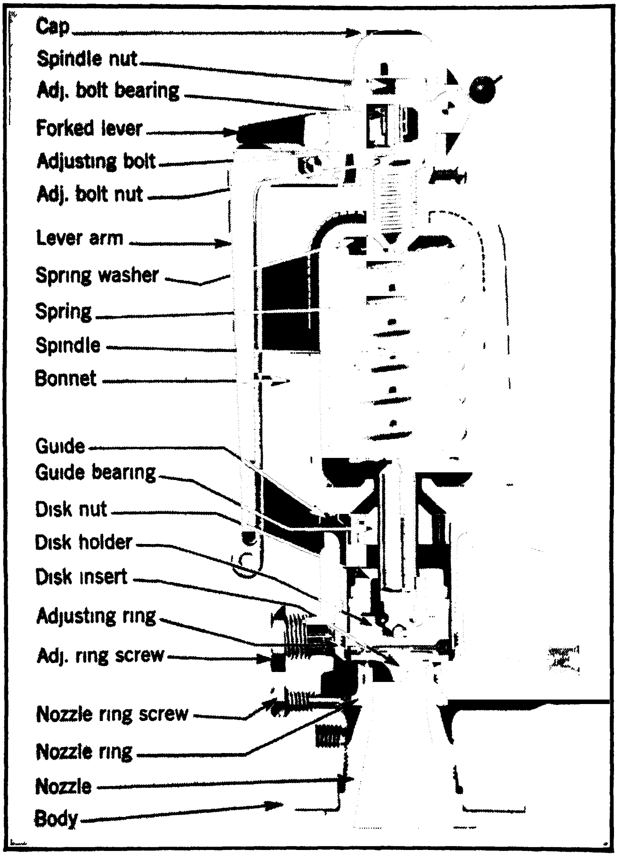


FIG. 68. Crosby pop safety valve.

methods of decreasing the formation of scale by purification of the feedwater are discussed in Chapter 8. The heat loss due to boiler scale varies with the temperature and rate of gas and water flow; experiments have shown a heat-conductivity loss of 12 per cent for 1/16-in. thickness of scale for boiler tubes (lower rows) near the furnace. Scale may be the cause of localized overheating, resulting in serious burning, blistering, and damaging of tubes and plates. All boilers not using distilled water require the use of a mechanical tube cleaner at definite intervals

in order to remove effectively the scale formed on the tube surfaces. *Mechanical tube cleaners* may be divided into two classes: (1) *vibrator type*, which loosens the scale by rapid blows and is applicable to fire- and water-tube boilers; and (2) *revolving cutter type*, which removes the scale by rotating cutting tools and is used in water tubes only.

As scale is accumulating on the water side of the boiler heating surfaces, soot is being deposited on the flue-gas side. Soot includes all the solid matter leaving the fuel bed with the gaseous products of combustion. Experiments have shown soot to be an excellent heat-insulating material, as evidenced by a heat-conductivity loss of 25 per cent for a soot thickness of 1/16 in. Boiler surfaces exposed to the flue gases are usually freed from ash and soot by hand-operated steam lances, mechanical soot blowers, brushes, or similar devices. *Mechanical soot blowers* are permanently installed in boilers and so located that all parts of the heating surface subjected to soot deposit may be cleaned with jets of steam, air, or a mixture of air and steam. The soot blower is designed to rotate in one direction by pulling an operating chain; the jets sweep over the blowing arc once every revolution.

Figure 63[3] shows the location of soot blowers as installed in generators of the Joppa Station, Art. 108. These automatic blowers are operated in sequential order, conforming with the flow of combustion products, to carry the fly ash out of the furnace. Compressed air is used for automatic control and also as the blowing medium. Full control of the operation is provided from a panel on the operating floor. High temperatures in some areas require that the soot-blower tubes be retracted from the furnace when they are not in use.

PROBLEMS

1. A water-tube boiler generates 16,000 lb of steam per hour at a pressure of 200 psia and a quality of 98 per cent when the feedwater is 75 F. Find (*a*) factor of evaporation, (*b*) equivalent evaporation, pounds per hour, (*c*) boiler horsepower developed, (*d*) per cent rating developed if the heating surface is 2000 sq ft, and (*e*) over-all efficiency if coal having a higher heating value of 9000 Btu per lb as fired is used at a rate of 3000 lb per hr.

2. A boiler produces dry saturated steam at 300 psia from feedwater at 180 F. Calculate the boiler horsepower necessary to supply a 100-hp engine if it requires 32 lb per hp-hr. 100.8 boiler hp, *Ans.*

3. A boiler evaporates 11 lb of water per lb of coal used under the following conditions: steam pressure, 200 psia; quality, 0.98; feedwater temperature, 190 F; and coal cost, $8.00 per ton. Find the cost per 1000 lb of steam from and at 212 F.

4. A steam generator produces 200,000 lb of steam per hr at 660 psia and 700 F. The feedwater enters the economizer at 300 F and leaves at 400 F. The steam leaves the boiler drum with a quality of 98 per cent. The unit consumes 22,500 lb per hr of

[3] Courtesy of Copes-Vulcan Division, Continental Foundry and Machine Company.

Pittsburgh County, Oklahoma, coal as received. Find, as percentages of the total heat supplied, the heat absorbed by (*a*) the economizer, (*b*) the boiler, and (*c*) the superheater. (*d*) Find the over-all efficiency of the unit.

5. The following data were obtained from a boiler test: water supplied to boiler, 60,000 lb per hr; feedwater temperature, 182 F; steam pressure, 200 psig; quality of steam, 99 per cent; and barometer reading, 28.55 in. mercury. (*a*) Find the kB absorbed per hour. (*b*) Determine the number of square feet of heating surface if the boiler develops 225 per cent of its rated horsepower during the test.

6. A small boiler is operated at 350 per cent of rated capacity. How many pounds of water are evaporated from and at 212 F per hr per sq ft of heating surface?

7. A power plant operated under the following conditions: coal used, 3200 lb per hr; higher heating value of coal, 13,000 Btu per lb as fired; water evaporated, 30,000 lb per hr; feedwater temperature, 180 F; boiler pressure, 156 psig; barometer, 14 psia; calorimeter pressure and temperature, 14.0 psia and 220 F; heating surface, 5000 sq ft. Find (*a*) kB absorbed per hour; (*b*) over-all efficiency of plant; (*c*) equivalent evaporation, pounds per hour; and (*d*) per cent rating developed.

8. The over-all efficiency of a steam generator is 82 per cent. The coal used has 12 per cent moisture and a higher heating value per pound (dry) of 12,000 Btu and is burned at the rate of 10 tons per hr as received. Steam pressure, 460 psia; total temperature, 600 F; and feedwater temperature, 202 F. (*a*) How many pounds of steam are generated per hour? (*b*) Determine the equivalent evaporation, pounds per pound of coal as fired. (*c*) If the coal is purchased at $7.50 per ton, find the coal cost per 1000 lb of equivalent evaporation.

9. A boiler generates 100,000 lb of steam per hr at 1000 F total temperature. Feedwater temperature, 294 F; steam pressure, 1200 psia; higher heating value of coal as fired, 11,000 Btu per lb; and over-all efficiency, 85 per cent. How many pounds of coal are required per hour? 13,200 lb per hr, *Ans.*

10. A boiler requires 19,000 lb of coal per hr; higher heating value of coal, 13,500 Btu per lb; temperature of feedwater, 200 F; steam pressure in boiler, 180 psig. A throttling calorimeter shows a temperature of 240 F at an atmospheric pressure of 28.53 in. of mercury. The over-all boiler efficiency is 76 per cent. (*a*) Find the kB absorbed per hour. (*b*) How many pounds of steam are generated per hour? (*c*) What is the quality of the steam produced?

11. Fuels of the following heat values are available for use: natural gas, 1000 Btu per cu ft; fuel oil, 19,000 Btu per lb; coal, 12,500 Btu per lb as fired; and lignite, 7000 Btu per lb as fired. The cost of natural gas is 45 cents per 1000 cu ft; fuel oil, $2.50 per 40-gal barrel (specific gravity, 0.87); coal, $7.00 per ton; and lignite, $4.00 per ton. The boiler efficiencies are 87 per cent for gas; 84 per cent for oil; 76 per cent for coal; and 60 per cent for lignite. Calculate the cost per 1000 lb of evaporation from and at 212 F for each fuel.

12. The steam used by a turbine is 12 lb per kw-hr at a pressure of 650 psia and a temperature of 700 F. The over-all boiler efficiency is 82 per cent with feedwater at 300 F. (*a*) How many pounds of 12,000-Btu coal are required per kilowatt-hour? (*b*) If coal costs $7.00 per ton, what is the fuel cost per kilowatt-hour?

(*a*) 1.315 lb per kw-hr, (*b*) 4.6 mills, *Ans.*

13. Calculate and tabulate a boiler heat balance based on the higher heating value of coal as fired, using the following data. Coal analysis, percentage by weight, as fired: C, 70; H_2, 6; O_2, 5; N_2, 2; S, 1; moisture, 4; and ash, 12. Higher heating value of coal as fired: 14,000 Btu per lb. Temperatures: feedwater, 180 F; flue gas,

460 F; and boiler room, 80 F. Steam is at a pressure of 300 psia and at a quality of 97 per cent. Combustible in refuse, 30 per cent; moisture per pound of dry air, 0.03 lb. Flue-gas analysis, percentage by volume, CO_2, 13; CO, 0.5; O_2, 4.7; and N_2, 81.8. The evaporation from and at 212 F is 11 lb per lb of coal as fired.

14. In a boiler test the following data were recorded. Weights: feedwater, 50,100 lb; coal used, 6280 lb; and dry refuse removed from ashpit, 900 lb (analysis showing 30 per cent combustible). Pressures: steam, 325 psia; and barometer, 29.7 in. Temperatures: feedwater, 175 F; steam, 600 F; flue gas, 550 F; and boiler room, 80 F. Percentage of saturation of room air, 55. Coal analysis, percentage by weight, as fired: C, 65; H_2, 5; N_2, 1; O_2, 8; S, 2; moisture, 8; ash, 11; and Btu per lb, 13,010. Flue-gas analysis, percentage by volume: CO_2, 13.3; CO, 0.55; O_2, 4.35; and N_2, 81.8. If the heating surface of the generator is 11,000 sq ft and the furnace volume, 2000 cu ft, find (*a*) the kB absorbed per hour; (*b*) the boiler horsepower developed; (*c*) the equivalent evaporation, pounds per hour; (*d*) the heat liberated per cubic foot of furnace volume; and (*e*) the over-all efficiency.

15. Using the data of problem 14, calculate and tabulate the complete boiler heat balance per pound of coal as fired. Show the method of calculation for each item.

16. In continuous operation a steam boiler is a steady-flow device. If water enters at a pressure of 200 psia, a temperature of 153 F with a specific volume of 0.01634 cu ft per lb, having internal energy of 120.85 Btu per lb, a velocity of 1200 ft per min, and an elevation of 5 ft, and leaves as steam at a pressure of 190 psia, a specific volume of 2.404 cu ft per lb, internal energy of 1113.1 Btu per lb, a velocity of 6000 ft per min, and an elevation of 60 ft, find the change in enthalpy per pound of feedwater 1076.45 Btu per lb, *Ans.*

CHAPTER 7

Steam Boiler Auxiliaries

113. Settings and Furnaces. The enclosing and supporting structure for a boiler is called the setting. Boilers having more than 1000 sq ft of heating surface are usually supported by steel frames independent of the brickwork because of the weight and the problems of expansion and contraction. The *furnace* is that part of the setting where fuel is burned.

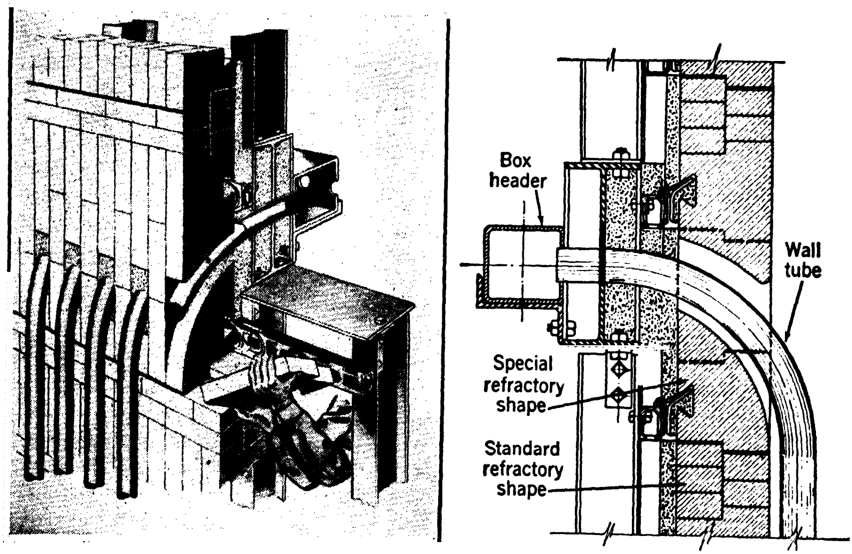


FIG. 69. Reintjes sectionally supported furnace wall.

Furnace walls are of three general types—solid refractory, supported or hung refractory, and waterwalls. *Solid refractory walls*, Fig. 52, are used for small, low-output boiler settings. The maximum furnace output may be limited by the point where the combined effect of load and temperature cause failure of the brick. Such a condition is called the

refractory end point and may be reached when an attempt is made to get maximum output from a given generator.

Supported or *hung walls* are carried by a steel framework to which special refractory shapes are attached. Figure 69 illustrates a wall of this type; one special shape of refractory supports a large number of

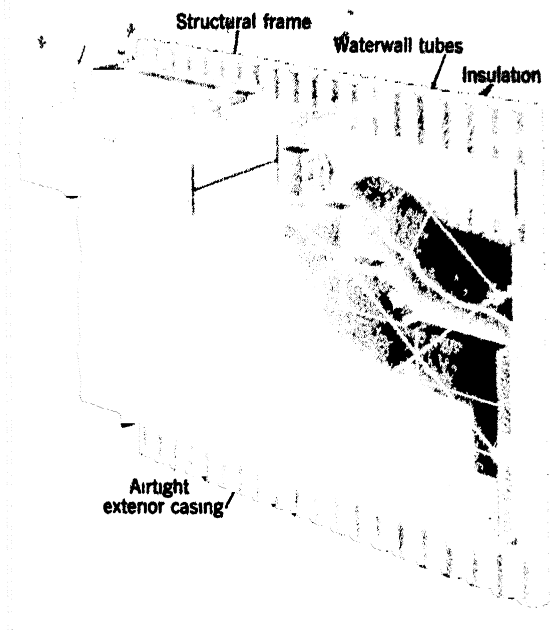


FIG. 70. Solid waterwall as used in B and W integral furnace boiler.

standard firebrick. The steel frame is cooled by air drawn in between the brickwork and the exterior sheet metal casing; this air becomes preheated and is used as secondary combustion air.

Waterwalls, Fig. 70, consist of a continuous row of tubes that are backed by refractory. The tubes protect the walls from erosion and are highly efficient in absorbing radiant energy. Very high steaming rates can be maintained without detrimental effects.

It is common practice to incorporate the furnace within the tube structure, thus using a minimum of brickwork. Such an arrangement, shown in Fig. 63, is called an *integral-furnace boiler*.

114. Grates and Stokers. A *grate* is a sectional metallic structure designed to support the fuel in a furnace and to permit passage of primary combustion air through openings to the fuel. Most grates are

made of sections of alloy cast iron since it is relatively inexpensive and is well suited to withstand high temperatures. The total area of the openings varies from 20 to 40 per cent of the grate area and is dependent on the kind of fuel used. For a given capacity, the total grate area needed depends on the heating value of the coal and the rate of combustion. Manufacturers usually provide about 1 sq ft of grate surface for every 40 to 60 sq ft of water-heating surface.

A *stoker* is a power-operated fuel-feeding mechanism and grate. It is used for supplying solid fuel to a furnace and admitting air to the fuel for the purpose of combustion. Several distinct types of stokers

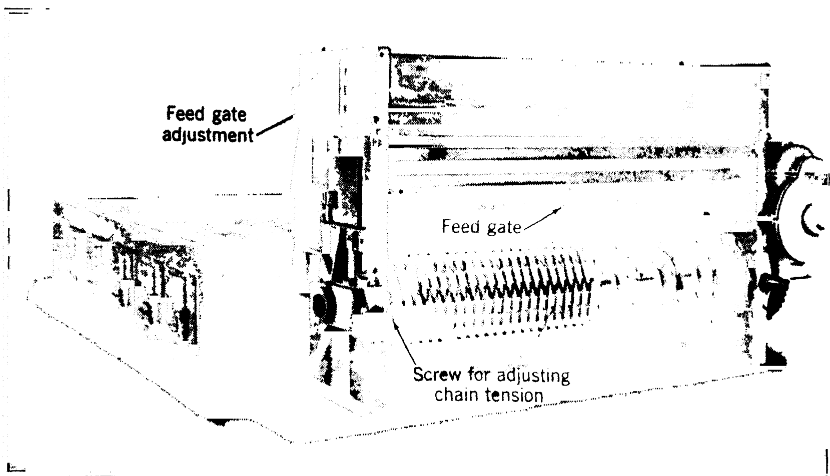


FIG. 71. Laclede forced-draft chain-grate stoker.

have been developed to suit special conditions such as size of steam boiler served, fluctuations in load, draft available, quality of fuel burned, and space limitations. By the use of stokers, a cheaper grade of fuel can be burned, a higher efficiency attained, greater flexibility of operation assured, better furnace conditions maintained, and less smoke produced than is usual with hand firing. Automatic stokers may be classified as *traveling-grate*, *overfeed*, *spreader*, and *underfeed*.

115. Traveling-Grate Stokers. One form of traveling-grate stoker, called a chain grate, consists of a belt of articulated links designed to operate over sprockets at the front and rear of the furnace, Fig. 71. The individual grate links are usually cast iron and are held together by rods that span the width of the grate. The links are so shaped that crushed coal feeding from a hopper at the front end of the furnace is adequately supported while primary air rises through or between the

links. The speed of the chain travel can be controlled manually or by automatic regulation to meet the need for steam production. The depth of coal layer used in the grate depends on the coal size and available draft and is controlled by a gate which extends across the front of the furnace and can be moved vertically. Fine coal resists the flow of air, whereas coarse coal allows air to pass through rather freely. Any segregation of sizes occurring on the grate will result in the coarse coal burning out first with consequent air leakage or the fine coal being carried over the end of the grate. The incoming coal is ignited by the

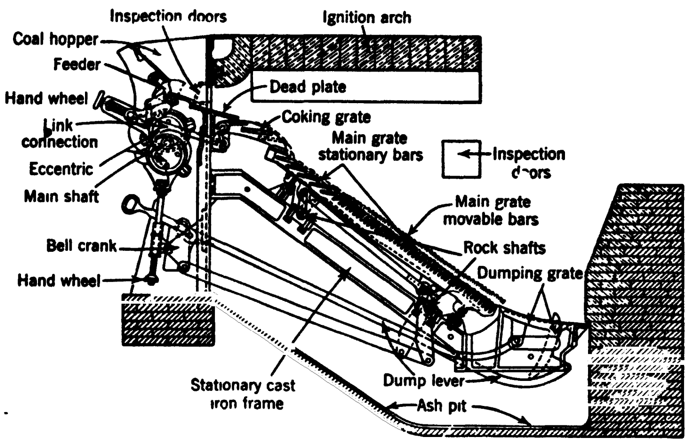


FIG. 72. Typical overfeed mechanical stoker.

heat reflected from an ignition arch similar to the arch shown in Fig. 72. The air flow to each section, or zone, of the stoker is controlled by dampers within or under the stoker. Combustion progresses as the grate moves to the rear and should be completed before the rear sprocket is reached because the residue is then dumped into the ash hopper. Zone dampers in the compartments of the stoker prevent the passage of excess air through grate areas where only a thin layer of ash remains.

Chain-grate stokers are widely used for burning bituminous coals which are noncaking and high in volatile matter and ash. They are also suitable for burning anthracite fines, coke breeze, and lignite. Air leaking through the ashpit and around the end of the stoker is directed over the grate by tubes called *water backs*, which support part of the bridge wall. These tubes also absorb the last energy available from the residue and prevent slag from forming on the bridge wall. Similar water tubes called clinker-chill tubes are sometimes placed horizontally in the side walls at the grate line to prevent clinkers from adhering.

A second traveling-grate design, Fig. 74, consists of a series of carrier bars spanning the furnace width and on which are fitted the castings that form the grate. All the grates may be removed without disturbing the carrier linkage.

With forced draft, stokers of the traveling-grate type are designed to burn from 40 to 60 lb of coal per sq ft per hr. With natural draft, the average combustion rate is about 30 lb of coal per sq ft per hr.

116. Overfeed Stokers. With stokers of the overfeed type, Fig. 72, the coal is pushed into the furnace automatically at the top of a sloping grate and fed downward by the movement of the grate bars and the aid of gravity. Refuse collects at the bottom where it is crushed by rolls and dumped. Overfeed stokers are adapted to the burning of nonvolatile caking bituminous coals which are easily ignited and which require agitation of the fuel bed. Arches of the sprung or suspended type are used for ignition and coking. This method of firing was widely used in the early development of mechanical stoking, and a considerable number of overfeed stokers are still in service in older steam plants where boiler units are not very large.

When overfeed stokers are arranged on each side of a furnace, they form a V-shaped receptacle at the base of which clinker grinders are usually installed. When used with natural draft, these stokers are not suitable for conditions where the load fluctuates rapidly. The maximum burning capacity is about 40 lb of coal per sq ft of grate area per hr with forced draft and about half that much with natural draft.

117. Spreader Stokers. Figure 73 illustrates a spreader stoker in which the coal is thrown into the furnace by the action of a rotating impeller. A plunger-type feeder automatically controls the rate of coal fed. *Primary air* for combustion is supplied to the furnace through the grate openings. *Overfire air* to create turbulence and supply oxygen for the fine coal which is burned in suspension is admitted through nozzles located directly above the arch. The grate surface is built in sections, each served by an individual rotor or feeder. When fires are cleaned, the fuel supply to one section of the grate is temporarily discontinued until the fuel on that section of the grate is consumed. The steam pressure is uniformly maintained by continuing the operation of the adjoining sections. The power-dumping grates are operated by steam or pneumatic cylinders, and the ash is dumped to a hopper.

Figure 74 shows a spreader stoker equipped with a traveling grate which discharges the ash continuously, thus avoiding periodic cleaning. High capacity can be maintained, and the unit can burn all types of solid fuels, including coking and noncoking coals. Coal particles are pre-oxidized as they pass through the high-temperature furnace gases,

and their tendency to coalesce is reduced Because of a uniformly thin fuel bed, low fan power is required and 1 in of water-gage pressure is

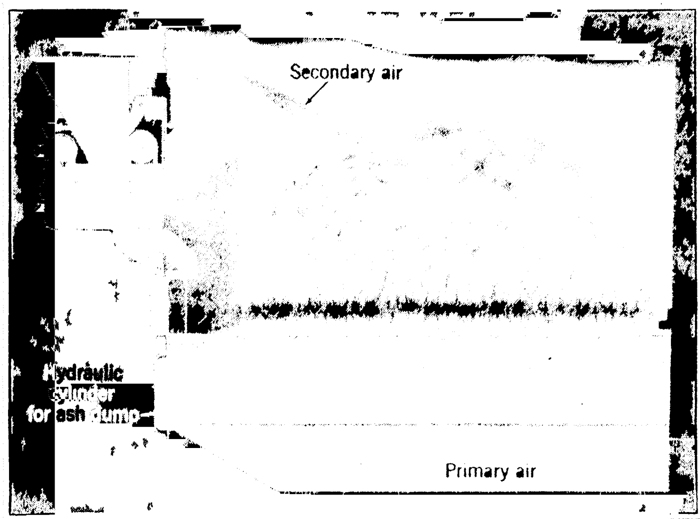


FIG 73 Erie City spreader stoker with dumping grate

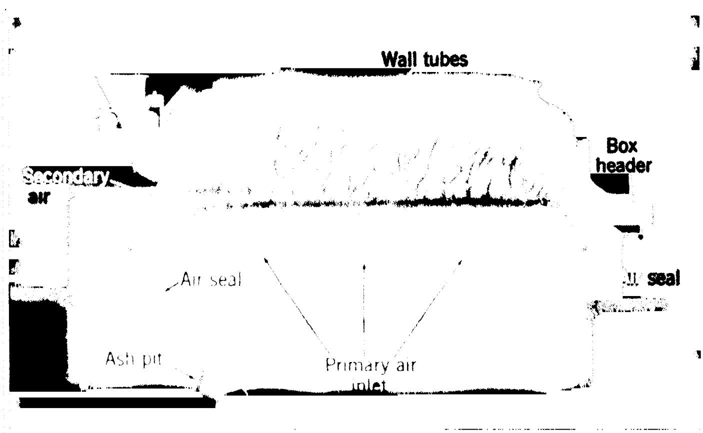


FIG 74 Erie City spreader stoker equipped with a traveling grate.

sufficient to maintain a combustion rate of 60 to 90 lb of coal per sq ft of grate area per hr in a stoker of this type.

118. Underfeed Stokers. In underfeed stokers, Fig. 75, the coal is introduced beneath the fuel bed. The smoke-producing volatile constituents distilled beneath the active combustion zone pass up through

the incandescent fuel bed and burn in a high-temperature zone under conditions conducive to rapid and complete combustion. Coal is pushed from the hopper into a retort by means of reciprocating plungers. The retort is trough-shaped, and the green coal is forced along the bottom. The fuel is gradually pushed up to the surface during the process of coking and ignition and feeds out to the grate. As the fuel bed is usually 24 in. or more in thickness, forced draft is essential, the air pressure below the fire being as high as 5 to 7 in. of water gage.

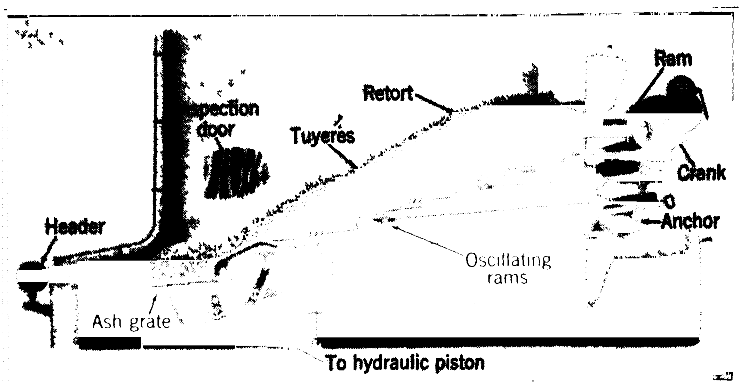


FIG. 75. American Engineering underfeed stoker.

Primary air is forced into the fuel bed through tuyères in the sides of the retort. Furnaces for underfeed stokers are not so large as others of equal coal-burning capacity, and they require no special arches to secure ignition.

Underfeed stokers are well adapted for nonclinkering high-volatile coals having caking characteristics and low ash content. The continual agitation of the fuel bed keeps the bed porous even with coals that cake. Eastern coals may be burned continuously at a rate of 60 lb per sq ft of grate surface per hr, and for short periods of time the combustion rate may be increased to 75 lb; for western coals, the combustion rate is lowered to approximately 50 lb per sq ft per hr.

119. Pulverized-Fuel Equipment. The use of pulverized coal in power plants has increased rapidly, and today more than half of the steam generated by coal-fired furnaces is being produced by pulverized fuel.

The two systems of preparing and burning pulverized fuels are (1) the central or storage system, with its independent preparation plant, transport system, bins, and feeders; and (2) the unit or direct-fired system in which one or more pulverizers prepare and deliver the fuel to the

furnace with sufficient air to suspend the dust and to burn the volatile matter. The first successful installations used the *storage system*, in which the pulverized fuel was stored in a bin to be fired as needed. Today, direct firing from the pulverizer is almost universally practiced because of its low cost, simplicity, and flexibility of operation.

The *direct-fired* (*unit*) *system*, Fig. 76, reduces the equipment to the minimum. A complete pulverized-fuel system is provided near each boiler, thus eliminating the need for storage and transportation facili-

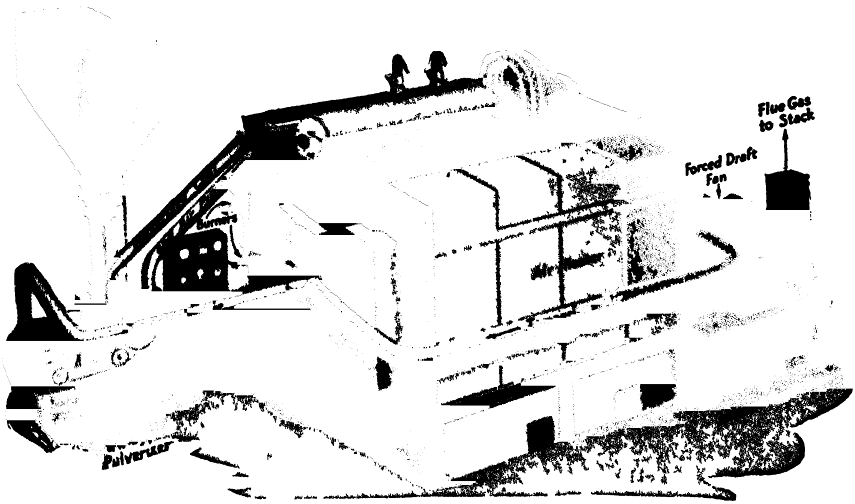


FIG 76 B and W direct-fired integral-furnace boiler.

ties. Preheated air forced through the pulverizer eliminates the necessity for separate driers. The fuel is first crushed to a general size of ¾-in. lumps and then passed over a magnetic separator to remove the tramp iron, such as bolts, nuts, and rivets. Next the crushed coal goes to the pulverizers and then directly to the furnace.

Pulverizers may be divided into four types, depending on the method of fuel reduction, namely, ball-race, bowl-mill, impact, and ball mills. The *ball-race mill* shown in Fig. 77 employs the spherical-ball and grinding-ring principle. The row of heavy steel balls is propelled (rolled) by the lower of the two grinding rings, driven as shown. The pressure between the balls and rings is applied and kept constant by externally controlled steel springs. Coal enters at the upper right side of the pulverizer; the coarser particles drop to the row of balls for pulverizing, while the finer particles are removed from the grinding zone by air passing between the balls. A stream of preheated air acts as both a

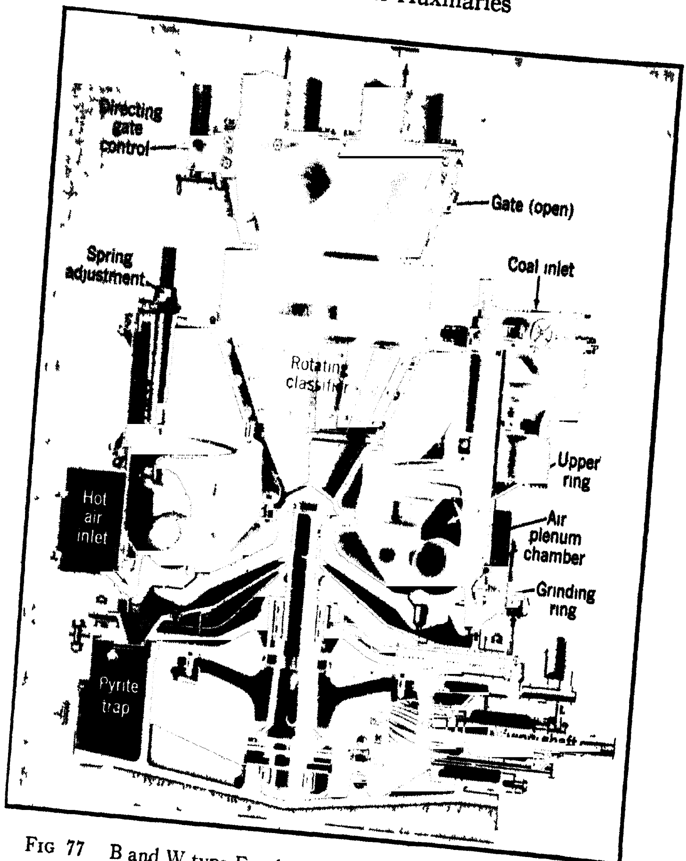


FIG 77 B and W type-E pulverizer, capacity to 15 tons per hour

drying and a separating medium. Separation of coarse material from fines is effected at the top of the pulverizer by the rotating classifier through which the air-coal stream passes. The coarse material returns to the grinding zone, and a homogeneous mixture of air and coal of the proper fineness passes to the burners through pipes attached to the top of the pulverizer. Pyrites and other hard materials pass through the pulverizer, Fig. 77, and are rejected as indicated without damaging the grinding elements.

The *bowl mill* grinds the coal between the inside of a whirling bowl and rolls mounted on pivoted axes, Fig. 78. Coal fed into the center is thrown by centrifugal force against the sides of the bowl where it is pulverized between the rolls and the grinding ring. Air coming up from the annular space around the bowl carries coal into a separator

above for classification. The coarse particles are thrown back to the bowl for further grinding, and the fine coal is carried to the furnace.

Impact pulverizers, Fig. 79, employ paddles, rings, or blades, which rotate within a casing at a speed of 1000 to 2000 rpm. The impact of the blades and the attrition of the fuel particles on each other effect the

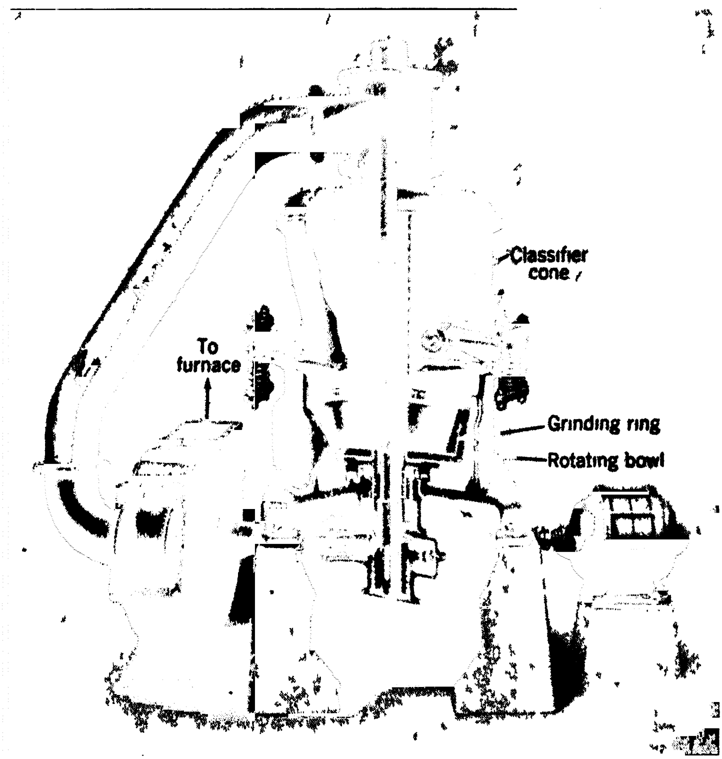


Fig 78 C-E Raymond bowl-mill pulverizer.

desired fineness. Impact-type mills are used for capacities up to 10 tons of fuel per hr

The *ball mill*, Fig 80, consists of a horizontal steel drum containing a large number of steel balls of various sizes In operation, the fuel enters one end of the drum and is crushed by the constant shifting of the balls as the drum rotates, preheated air is the drying and separating medium.

Coal is fed to pulverizers at a desired rate, and after being pulverized it is removed by the flow of air produced by a fan which sometimes is integral with the pulverizing mechanism. In direct-fired installations

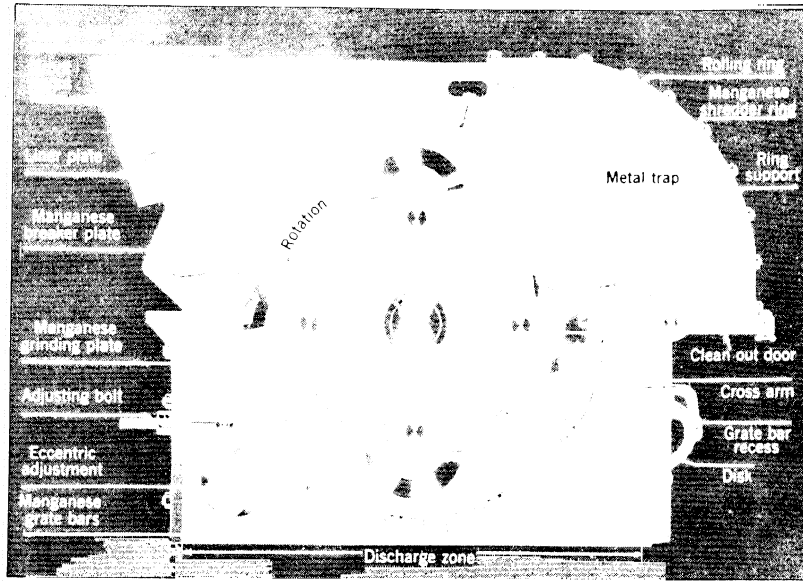


FIG. 79. American impact pulverizer.

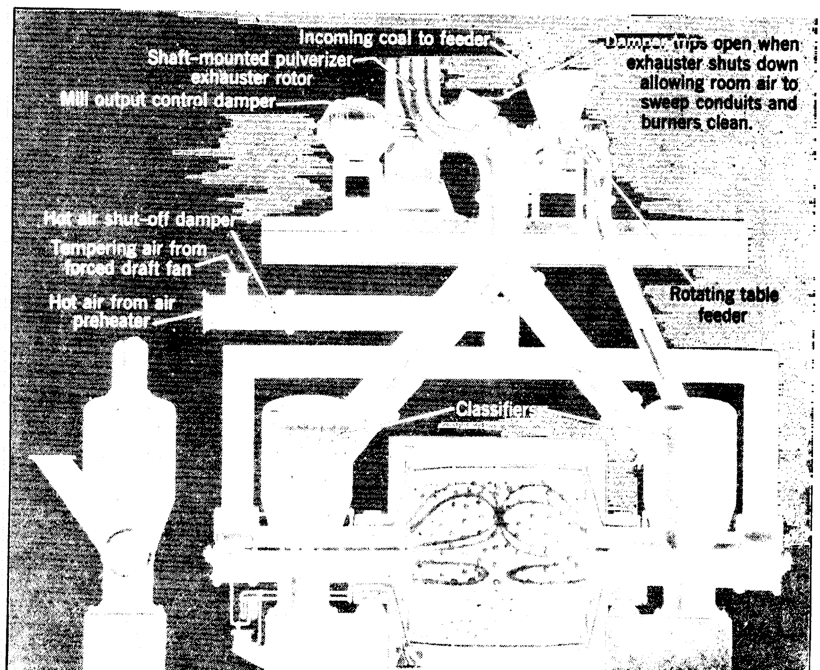


FIG. 80. Foster Wheeler ball-mill pulverizer, showing the feeder, ball mill, double classifier, and exhauster.

the primary air supplied to the furnace with the fuel is that air which passes through the mill. The degree of fineness of the discharged fuel depends largely on the velocity of the air passing through the mill. A fineness of 75 to 80 per cent through a 200-mesh (openings per inch) and of 100 per cent through a 50-mesh sieve is considered necessary in most cases to prevent slagging troubles and to insure complete com-

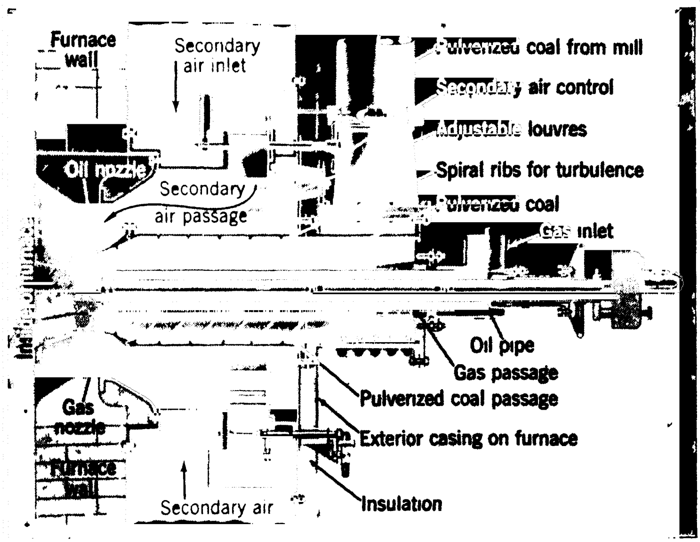


FIG. 81. C-E burner for pulverized coal, oil, or natural gas.

bustion. Some authorities claim that uniform fineness is more important than extreme fineness.

Pulverized-coal burners fire finely divided coal suspended in the primary air, the secondary air being admitted around the burner, below the burner, or elsewhere in the furnace. High-volatile coal ignites more quickly and burns faster and more completely than low-volatile. High- and medium-volatile coals are generally fired through horizontal burners, Fig. 81, with practically all the secondary air supplied around the burners. Low-volatile coal requires a rich mixture in the burner with very little primary air; hence contact with the secondary air is delayed. With such fuel it is customary to use vertical burners and fire downward through an arch with some of the secondary air supplied around the burners and the remainder through the furnace walls.

Turbulence may be increased by burner arrangement as well as by burner design. In *opposed firing*, two burners in opposite furnace walls fire directly against each other to produce more intimate mixing of the

air and the pulverized fuel; the *cross-firing* method is a combination of vertical and horizontal firing with intersecting coal streams. *Tangential-firing* burners stacked above each other in the four corners of the fur-

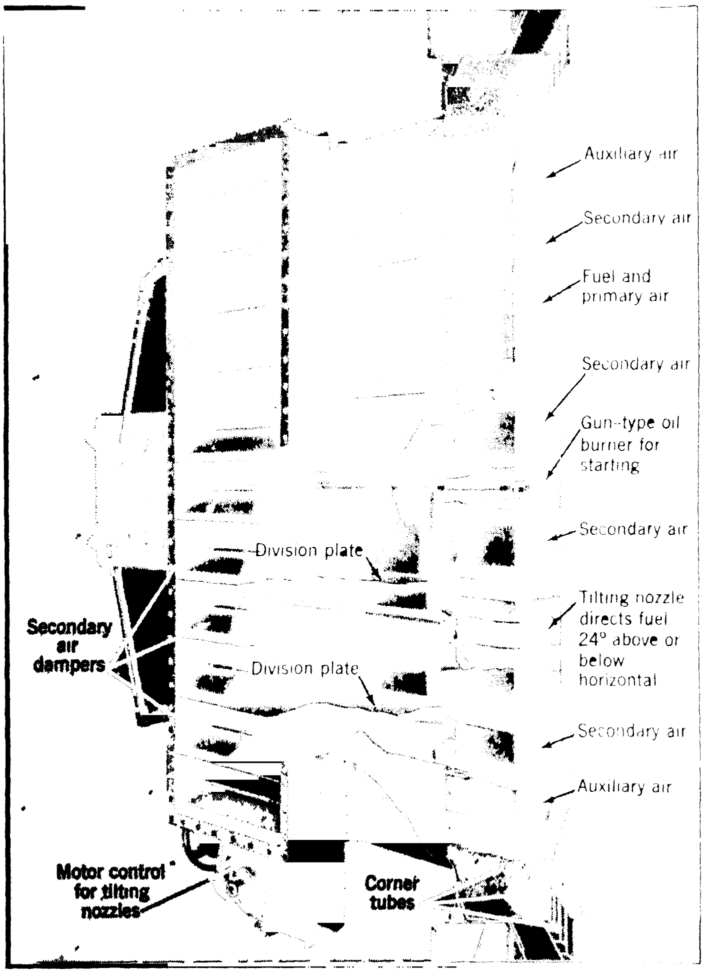


FIG 82 C-E burner for tangential firing.

nace send horizontal streams of air and coal tangent to a circle whose diameter ranges from 2 to 4 ft. Figure 82 shows a pulverized-coal burner arranged for tangential firing. All air and fuel nozzles can be tilted 24 deg above and below horizontal.

Advantages of pulverized coal include complete combustion, simpler ash disposal, absence of smoke, use of cheaper coal, equipment adapt-

able to other fuels—oil or gas—easy control of fuel and air supply, absence of all moving parts in the furnace, no stand-by losses, low labor

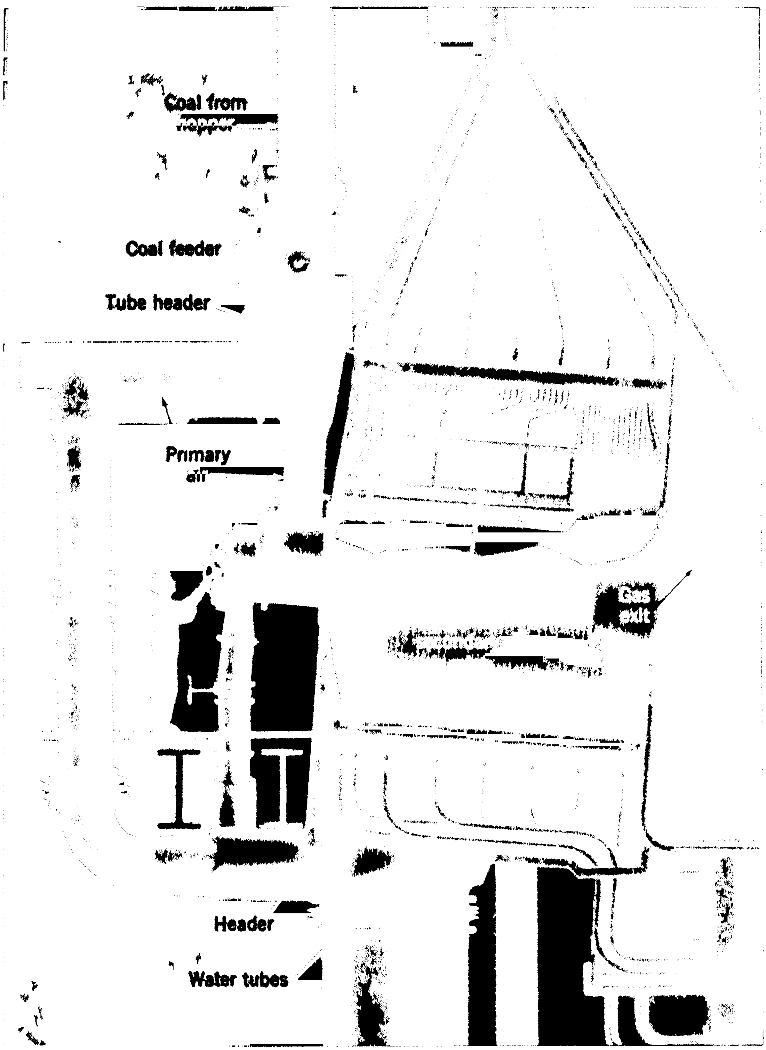


FIG. 83. B and W cyclone furnace.

cost, increase of capacity per unit of furnace volume, and higher efficiency. Some of the disadvantages are high first cost, cost of fuel preparation, furnace depreciation, and the possibility of fly ash discharging through the chimney.

120. Cyclone Furnaces. Figure 83 shows a cross section of a cyclone furnace which receives its name from the nature of the flow within the burner. The air and crushed coal are brought in tangentially to the circular burner in such a manner that violent rotation and mixing are accomplished. The coal is thrown to the outside of this whirlpool and imbeds in the molten slag film, where it burns as it is swept by secondary air at high velocity. As the slag melts it drains off and is allowed to fall into a slag-quench tank from which it can be sluiced out. By centrifugal action most of the fly ash is slung against the walls of the furnace where it is trapped by the molten film of slag and drained away. Thus clean stack gases flow from the burner to traverse the rest of the heat-exchanger passages. Better heat transfer is thus obtained, the fly-ash nuisance is reduced, good combustion is obtained, and less furnace cleaning is required.

121. Oil- and Gas-Burning Equipment. A burner designed for oil, gas, or pulverized coal is illustrated in Fig. 81. Oil burners should atomize the oil and thoroughly mix it with the air supplied for combustion. When these two functions are accomplished complete combustion can be obtained with the minimum of excess air. Atomization may be effected by air, gas, or steam at pressures from 25 to 100 psig or by mechanical means. Although compressed air can serve for both combustion and atomization, its use is restricted to small units on account of the high cost of air compression. Natural gas and by-product gases sometimes serve as atomizing agents at pressures of 50 lb or higher. The gas used for atomization assists in maintaining ignition in burning special fuels.

Steam is an effective but uneconomical atomizing agent. It is possible under the best conditions to operate steam atomizers with a steam consumption of 0.10 to 0.20 lb of steam per lb of oil fired. The steam can be taken directly from the boiler which is being fired; however, such a system is not self-starting for a single boiler, and other means of oil atomization are necessary until steam pressure has been developed. Other disadvantages of steam for atomization are loss of boiler water, noise of operation, frequent cleaning of burners, extra heat loss, and increased moisture in the stack gases. Steam atomizers are seldom designed to pass more than 1000 gal of oil per hr.

Mechanical atomization is accomplished by pressurizing the oil by means of a positive displacement pump or by centrifugally dispersing the oil from a rotating wheel. The former is called a *gun-type burner*, and the latter a *centrifugal burner*. The gun-type burner is lower in initial cost and requires less maintenance since there are no moving parts. Oil is supplied to the gun-type burner at pressures of 200 to

250 psig. A supply line carries excess and circulated oil back to storage or to a heater. Concentric pipes carry the entering oil to a point near the nozzle end at which some oil enters the furnace and the rest returns through one of the concentric pipes to the return header. By

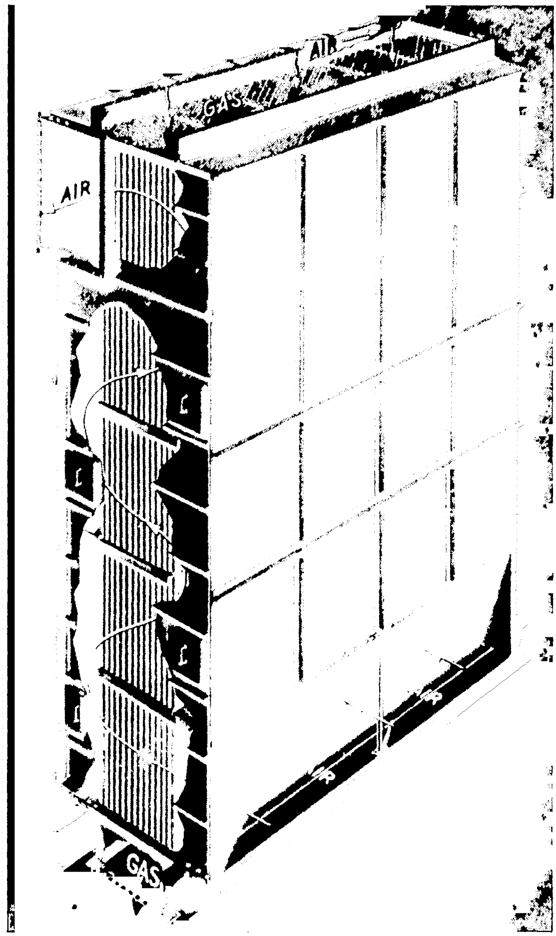


FIG. 84. Cutaway of C-E tubular air heater.

connecting all the oil return lines to a common return header provided with a master control valve, the return pressure can control the capacity of all burners simultaneously. Single mechanical oil burners have satisfactorily handled as much as 5000 gal per hr.

Burners for firing gas are designed for low-pressure gas (1/8 to 4 psig)

and for using gas directly from transportation lines where the pressure ranges from 30 to 50 psig In the latter case automatic regulators are used to reduce the pressure to a range of 5 to 25 psig

122. Air Heaters. Extracting waste heat from flue gases and transferring it to the combustion air requires air heaters. Advantages de-

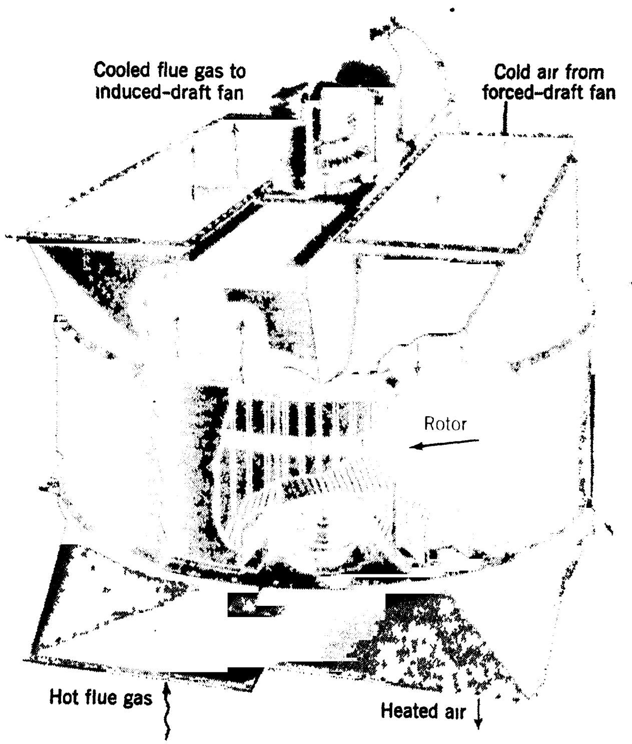


FIG. 85. Ljungstrom (Air Preheater) regenerative-type air heater.

rived from the use of preheated air for combustion are energy conservation, improved combustion, successful burning of low-grade fuel, increased efficiency, and increased capacity The hotter air supply increases the furnace temperature and hence increases the transmission of radiant heat to the boiler.

Air heaters apply either the *convection* or the *regeneration* principle of heat transmission The *convection* type may be subdivided as *tubular* or *plate.* Figure 84 shows a sectional view of a tubular-type air heater

in which counterflow occurs. The hot gases enter the tubes at the top and exit at the bottom. The cold air enters at the bottom right side, flows across the outside of the tubes, and leaves as indicated. Horizontal baffles direct the flow of air so that maximum heat transfer is obtained. Baffles are normally arranged to keep the velocity of the air as nearly constant as possible. In the plate-type heater the gas and air travel through alternate passages between the plates.

The *regenerative* air heater is illustrated in Fig. 85. It consists of a drum-type metal labyrinth which is rotated slowly, being alternately heated by the gases and cooled by the air. Stack gases flow upward through half of the labyrinth, and air, from a forced-draft fan, flows downward through the other half. A dividing seal is provided to prevent mixing of the air and gas as much as possible. The metal labyrinth is designed to provide the maximum surface area and the minimum flow resistance to the air and gases.

123. Superheaters. Superheaters are simple heat exchangers for imparting additional energy to steam above that contained in the saturated state for a given pressure. Modern high-pressure boilers deliver steam at the superheater outlet with temperatures as high as 1150 F. Superheaters receiving their energy from the stack gases are known as the *convection* type; those exposed to the radiant energy of the flames are called *radiant* superheaters.

The problem of maintaining a constant maximum steam temperature while varying the capacity of the boiler has led to a variety of superheater arrangements and locations within the furnace. A constant superheat is desirable in a given installation, particularly one of high pressure, since changing the steam temperature above or below the design value may result in losses.

Convection superheaters deliver a higher superheat as the boiler output is increased; radiant superheaters deliver a lower superheat. When properly sized and arranged in series the two tend to give a constant steam temperature, regardless of changes in steam flow rates. Other schemes for controlling superheat are (1) separately fired superheaters, Fig. 60, (2) adjustable burner heads which may be directed towards or away from the superheater, Fig. 82, (3) by-pass dampers that may direct gases across or around the superheater elements as needed, (4) injection of boiler feedwater as a spray, Fig. 57, (5) passing some steam through a submerged coil in the drum, or (6) using a closed heat exchanger to control the superheat by the incoming feedwater. The last three types are called *attemperators*. The convection type of superheater may be divided, as in Fig. 86, with the saturated steam passing first through the *primary section*, then through the *secondary section* in

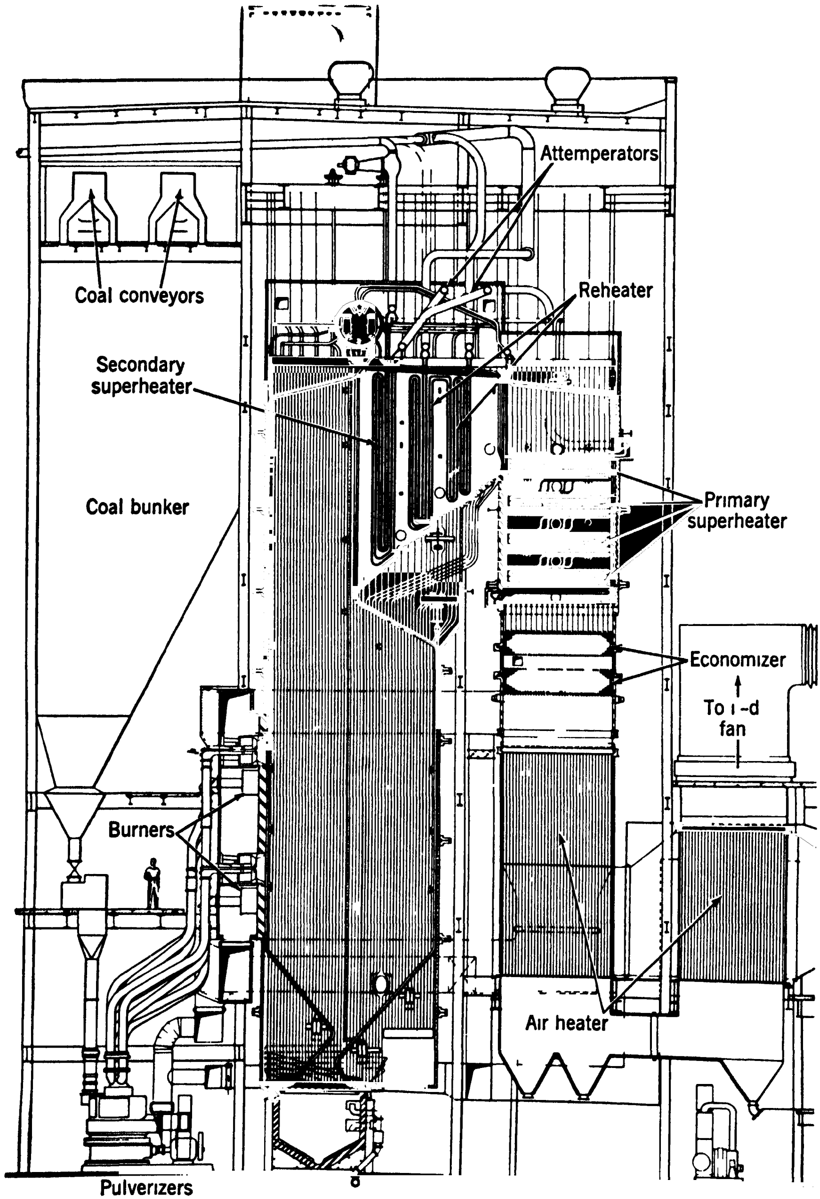


FIG 86 B and W central station boiler with primary and secondary superheaters

counterflow to the stack gases with a spray-type attemperator installed between the two.

A *reheater* is essentially a superheater as it is designed to bring the partially expanded steam back to superheat temperature by passing it through tubes placed in the gas stream. Figure 86 also shows a reheater mounted between the primary and secondary superheaters and equipped with a spray-type attemperator installed in the steam line ahead of the reheater. Temperature-operated automatic controls regulate the water spray to give constant reheat temperature.

124. Desuperheaters. Saturated instead of superheated steam is required by many industrial processes—the manufacture of sugar, paint, oil, textiles, cellulose, paper, and certain processes in connection with brewing, distilling, bleaching, and dyeing. If, in such plants, there is a demand also for power which can be generated more economically by the use of superheated steam, it is necessary to provide both superheated and saturated steam. The use of separate boilers is complicated; therefore, practically all modern industrial plants superheat all the steam and, by means of a desuperheater, remove the superheat from that portion required for certain equipment and processes.

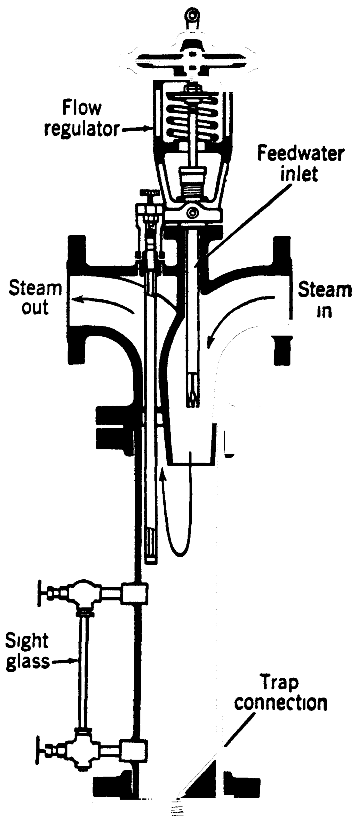


FIG. 87. Copes-Vulcan spray-type desuperheater.

Desuperheaters are either spray type, Figs. 87 and 88, or surface type. The former removes the superheat by the injection of boiler feedwater. In the latter the superheat is removed through a heat exchanger in which the steam and water do not mix. Each type is subject to automatic regulation.

125. Steam Purification. As steam rises from the lower tubes in a boiler it carries with it water, foam, and suspended solids. The term *steam purification* is applied in general to a process which is used to prevent foam and solid particles from being carried away from the drum with the saturated steam. The apparatus usually consists of an inte-

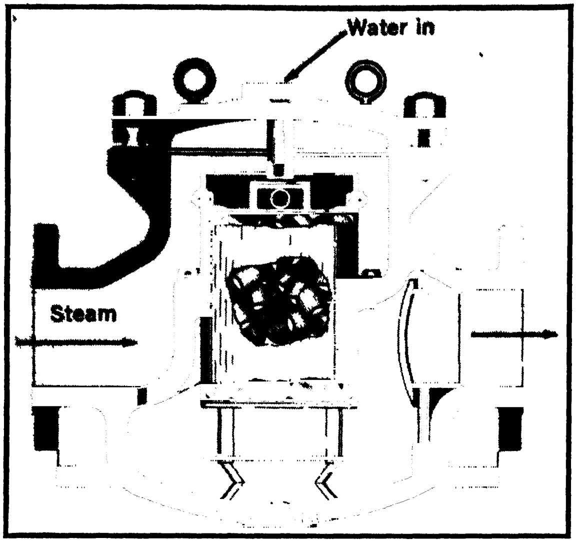


FIG. 88. Schutte and Koerting absorption-spray-type desuperheater.

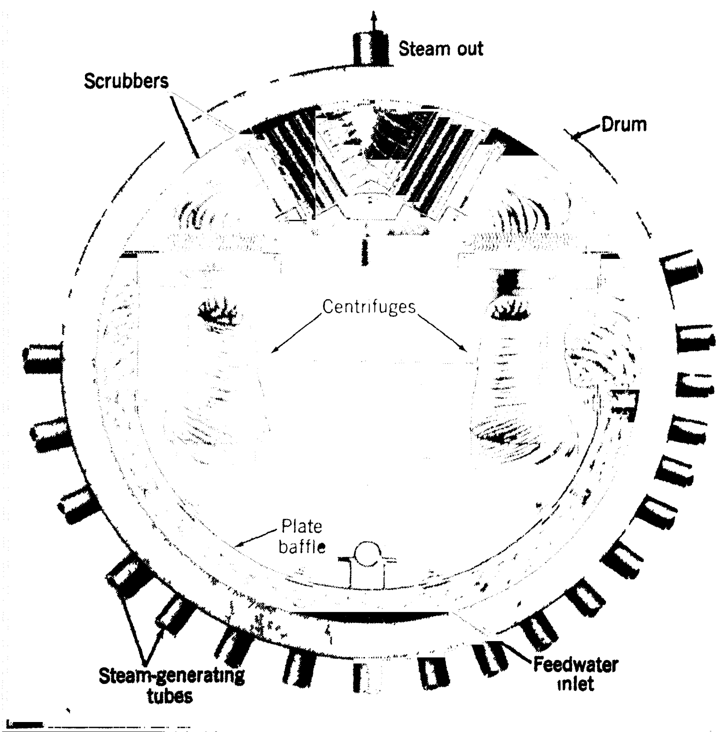


FIG. 89. B and W cyclone steam separator.

grated combination of parts, as illustrated in Fig. 89, in which the steam and water first pass through a centrifugal separator where the water is directed back to the drum and the steam rises through corrugated baffles called dryers or scrubbers. Fine droplets of moisture striking the scrubber tend to adhere and drain back to the drum. Solid particles that are carried in suspension are therefore prevented from being drawn into the superheater tubes, where complete evaporation would cause them to be deposited as scale. Boiler tubes are more accessible for cleaning than superheater tubes and can be *blown down;* i.e., a valve may be opened at the lowest point in the boiler, allowing water and suspended solids to escape.

126. Ash Disposal. Modern power plants make use of various types of ash-disposal systems, such as (1) hydraulic sluice, (2) pneumatic conveyor, (3) chain or bucket conveyor, (4) motor truck, or (5) track-operated car. The choice of a system is a matter of economy and convenience. About 50 per cent of the ash from pulverized coal is carried through the boiler passes and into the stack, unless fly-ash separators are used. Fly ash can be collected by electric precipitation, water sprays, and cyclone collectors, Fig. 57. Electric precipitation is the most effective but it is expensive.

In *slag-tap furnaces* the ash is drawn off in a molten state, quenched and broken up by water jets, and carried away by *sluicing*. This method is especially applicable where the ash from the fuel has a low fusing temperature, below 2600 F. Solid or granular ash can be conveniently sluiced to a pit from which it can be transferred to cars by a clamshell dipper.

Pneumatic conveyors are convenient for removing ash and residue from the various points where it accumulates about a furnace setting—ashpit, dust separators, and drip hoppers at the front and below the stoker. The system is comprised of thick-wall cast-iron pipe extending from the point of ash pickup to a centrifugal separator mounted above an ash storage bin. An air pipe from the top of the centrifugal separator is connected to a steam-operated ejector at the base of the stack. When ash is being removed, the ejector induces a vacuum throughout the conveyor system. Opening an ash gate at any point in the system allows air to enter and flow at high vlocity through the separator to the stack. Any ash entering the gate is carried by the air to the separator where it is thrown out by centrifugal force and dumped to the hopper below.

Bucket and *drag conveyors* are widely used in small plants as a means of conveying ash to a hopper from which it may be gravity-loaded to freight cars or trucks.

127. Combustion Controls. Automatic devices for maintaining the desired steam pressure and the proper fuel-air ratio against changing load demands are called combustion controls. Formerly only a pressure gage was available, and the fireman anticipated the fuel needs by watching the change in pressure. Much excess air entered the furnace while the firing door was open, and a decrease in load could not be rec-

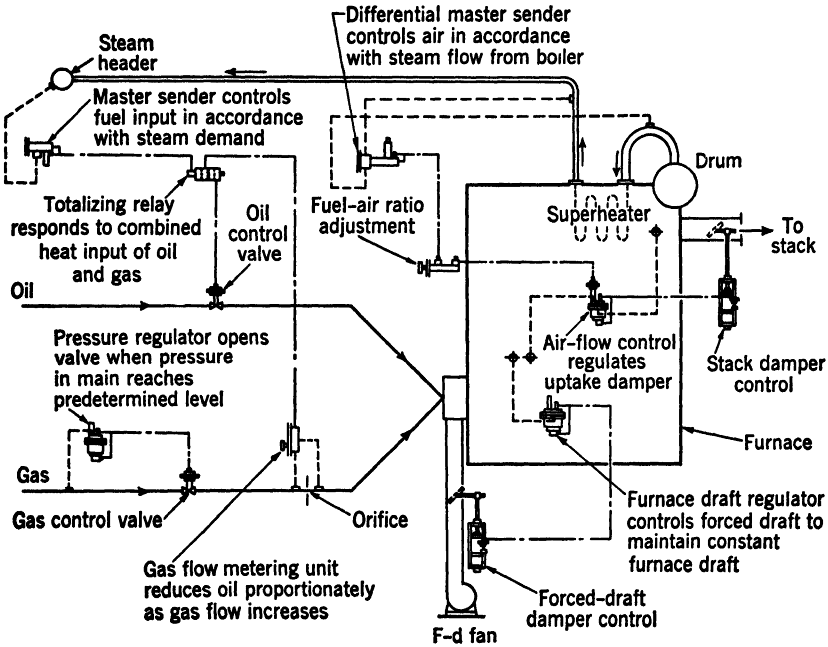


FIG. 90. Hagan boiler control system for gas or oil or both gas and oil.

ognized in time to prevent the boiler from venting to the atmosphere through safety valves.

Automatic controls use pneumatic, hydraulic, or electric power to control servomotors, which in turn adjust the fuel feed and the air supply quickly and simultaneously in response to changing steam demand. Three basic types of controls are in use: (1) on-off systems, (2) positioning systems, and (3) metering systems.

The *on-off system*, frequently found on domestic heating plants, is used on small boilers fired with oil or gas. The system operates at full capacity until a thermostat or pressure gage shuts off the fuel and air. With a given drop in temperature or pressure the flow of fuel and air is again started, and continues at a fixed setting. A "hold-fire" device

may be used with stoker operation to maintain a fire during the off period.

The *positioning system* consists of a pressure-responsive element that transmits an impulse to a servomotor which adjusts the speed or damper setting on the fuel and air supply motors. The system must be balanced so that a given drop in steam pressure produces a proportionate increase in fuel and air flow.

The *metering system* measures the steam flow and balances it against the impulses calling for more or less flow. This system is more sensitive than the positioning system since it does not rely on a given movement of a damper to produce a given increase in flow, but adjusts according to the flow itself. It is therefore not affected by voltage variations and changes in air pressure and temperature.

Figure 90 shows a pneumatic system in which the stack damper is controlled by the rate of steam flow, the fuel flow by the steam pressure, the forced-draft fan damper by the furnace pressure. The three circuits work independently but simultaneously. Any desired fuel-air ratio may be maintained.

PROBLEMS

1. Steam enters a superheater at a pressure of 300 psia and a quality of 98 per cent. It leaves the superheater at a pressure of 297.5 lb and a total temperature of 600 F. Determine the heat added per pound of steam in the superheater.

2. Determine the pounds of feedwater at 180 F which will be required to remove all the superheat from 1 lb of steam at 600 psia and 650 F in a spray-type desuperheater. 0.1119 lb per lb, *Ans.*

3. A surface-type desuperheater receives steam at 400 psia and 580 total temperature. Ten thousand pounds of steam per hour are to be desuperheated by 5000 lb of feedwater entering at 180 F. Determine the exit temperature of the feedwater.

4. A convection-type superheater receives steam at 450 psia and 98.5 per cent quality. The steam leaves the superheater at 720 F. Find (*a*) enthalpy increase in the superheater, Btu per pound; (*b*) specific-volume increase, cubic feet per pound; and (*c*) mean specific heat of the superheated steam, Btu per (pound) (degree F).

5. A chain-grate stoker 9.5 ft wide receives crushed coal having a mean specific weight of 54 lb per cu ft. The coal is spread to a uniform depth of 6.8 in. by the coal gate. Determine the stoker speed in feet per minute if the furnace is to burn 7.5 tons per hr. 0.86 fpm, *Ans.*

6. A boiler delivers 50,000 lb of steam per hr having a quality of 0.97. Another boiler delivers 30,000 lb per hr of steam having a total temperature of 500 F. Both boilers operate at 300 psia and deliver steam to the same header. Assuming thermal equilibrium, what is the temperature of the steam in the header?

7. Air is forced through a preheater at the rate of 180,000 lb per hr and heated from 80 to 350 F; flue gases at a temperature of 600 F enter the heater at the rate of 188,000 lb per hr. Find the exit temperature of the stack gases if c_p for air and gases is 0.25 Btu per (lb) (deg F).

8. A single-pass counterflow air preheater is designed to cool 900 lb of stack gases per min from 600 F to 400 F. Air-inlet temperature is 100 F, and the flow is 860 lb per min. Specific heat of air, 0.24; and of gases, 0.25. Neglecting radiation, find exit temperature of the air.

9. Coal at 80 F and containing 15 per cent moisture by weight is to be dried by 500 F air. Two and one-half pounds of air are circulated through the pulverizer per pound of coal. Assume that the water evaporates at a partial pressure of 1 psia, c_p of air is 0.24, and c_p of coal, 0.7. Determine the temperature of the dried coal and air leaving the mill.

10. An oil-fired superheater with an air-fuel ratio of 20 to 1 (lb per lb) sends flue gases at 1600 F across a convection-type superheater. The gases are reduced to 1300 F. Twenty pounds of saturated steam at 250 psia are superheated per pound of oil fired. Determine the superheat in degrees F added to the steam. Mean specific heat of gases, 0.26.

11. A steam generator burning 800 lb of oil ($C_{12}H_{26}$) per hr with 10 per cent excess air sends the products of combustion across a convection superheater. The gases are cooled from 1700 F to 1240 F. Determine the energy absorbed by the superheater, Btu per hour. Mean specific heat of gases, 0.26.

12. Radiant heat is transferred from a luminous flame to a black body by the equation $Q = 0.173 \times 10^{-8} \times A(T_1^4 - T_2^4)$, where A = area, sq ft; T_1 = flame temperature, deg F abs; T_2 = tube surface temperature, deg F abs; Q = heat, Btu per hr. Calculate the energy absorbed per hour by 1 sq ft of radiant superheater surface if the temperature of the flame is 3200 F and that of the superheater surface is 560 F and the superheater surface is assumed to be a black body (perfect absorber of radiant heat). 308,600 Btu, *Ans.*

13. Soot blowers operated at 8-hr intervals decrease the temperature of the stack gases 60 F. If 20 tons of coal are burned per hr with 12 lb of stack gases produced per lb of coal and if the temperature of the stack gases at the time of blowing is 540 F, determine the saving per 24-hr period. Assume mean specific heat of the gases to be 0.26 and increase in stack temperature to be linear with time.

14. An air heater is installed in the uptake of the stack of a cement mill to utilize part of the waste heat in the gases from the cement kiln, and the air passing through absorbs 10 per cent of the heat supplied bv the fuel in the kiln. Twenty thousand cubic feet of natural gas are burned per hour, and 13 2 cu ft of air are supplied per cu ft of gas. The gas has a heating value of 1000 Btu per cu ft, and the air has an initial temperature of 75 F. Calculate the final temperature of the heated air. Average density of flue gases is 0.04, of air is 0.05, and of natural gas is 0.07 lb per cu ft. Use $c_p = 0.25$ for air.

CHAPTER 8

Feedwater Heating and Conditioning

128. Feedwater Heating. The economical production of power necessitates the utilization of a maximum amount of the heat that may be obtained from a fuel. In modern power plants, heat that would be lost in either exhaust steam, from auxiliary equipment, or in flue gases, escaping to the stack, may be profitably used to heat boiler feedwater. Exhaust and bled steam are used in feedwater heaters; economizers serve as heat-transfer equipment in connection with flue gases.

The distinct advantages that may accrue from feedwater heating are: (1) reduction of boiler strains; (2) utilization of heat otherwise likely to be lost; (3) partial purification of untreated feedwater; (4) closer approximation of ideal thermal efficiencies for the plant by bleeding steam from turbine stages; (5) relief of the last stages of steam turbines from great volumes of low-pressure steam by stage bleeding to feedwater heaters; and (6) increased boiler capacity.

129. Economy of Feedwater Heating. Many boilers do not show an increase of efficiency when supplied with heated water; others show an efficiency increase or decrease due to the change in the rate of heat transfer. In any event, the saving due to feedwater heating appears in the reduced consumption of fuel and the increase in over-all efficiency for the plant. The amount of the fuel saving is dependent upon the initial and final temperatures of the feedwater and the total heat content of the steam produced, where waste heat is used for water heating. The saving effected is a variable dependent upon the operating conditions of the plant.

$$S = \frac{(h_{f,2} - h_{f,1})100}{(h_g - h_{f,1})} \tag{184}$$

where S = fuel saving, percentage.

h_g = total enthalpy of the boiler steam, Btu per lb.

$h_{f,1}$ = initial enthalpy of liquid of the feedwater, Btu per lb.

$h_{f,2}$ = final enthalpy of liquid of the feedwater, Btu per lb.

130. Feedwater Heater Classifications. When the heat supply is from hot flue or waste gases, a water heater is designated as an *economizer*. When the heat utilized is derived from exhaust, bleeder, gland, jet, or live-steam sources, the term *feedwater heater* is applied to the equipment. Feedwater heaters of the *direct-contact* or *open type* utilize the heat of the steam by direct contact of the steam and water as they are mixed. Direct-contact heaters operate at pressures ranging from slightly less than atmospheric to 30 psig. The steam and the water are at the same pressure. In the *closed* or *surface type* of heater the heat of the steam is transmitted through metal walls, and the steam and the water are not in direct contact. Closed heaters may operate at any steam pressure ranging from less than atmospheric to high. In closed heaters the water and the steam pressures are not usually the same.

131. Direct-Contact Feedwater Heaters. These heaters may be built as either vertical or horizontal units. Either form of construction requires the following principal parts: an enclosing shell of either cast iron or steel, water control, drip trap, blow-off connections, steam or air vent, and possibly a steam and oil separator at the steam inlet. The construction of direct-contact feedwater heaters may involve a tray or pan system over which the water overflows, or cascades or water spray nozzles may be used to bring the water into contact with the heating steam. In recent years stainless steel has been used as a material of construction for internal parts of heaters which are subjected to the actions of corrosive gases.

The vertical deaerating direct-contact heater of Fig. 91 operates to eliminate air and other gases carried by feedwater. The incoming feedwater makes two passes through the U-shaped tubes of the vent condenser where vapor coming from the top of the shell of the unit is reduced to water. This liquid is returned to the heater through a pipe connection fitted with a water-seal loop and a discharge spray. The noncondensable gases escape from the condenser shell through a vent pipe leading from it. The feedwater flows from the condenser tubes to a distributing box which has serrated edges to aid in properly directing it to the top bank of staggered trays. Here initial heating of the water takes place. Water overflows from the top heating trays to a lower bank where further heating and liberation of gases occur as the temperature of the water rises to that of the incoming steam. In general, within the heater shell, counterflow of the water and the steam exists. The distributing box and all trays may be of either corrosion-resisting cast iron or stainless steel.

Direct-contact feedwater heaters fitted with internal vent condensers may require less installation head room and less condenser maintenance

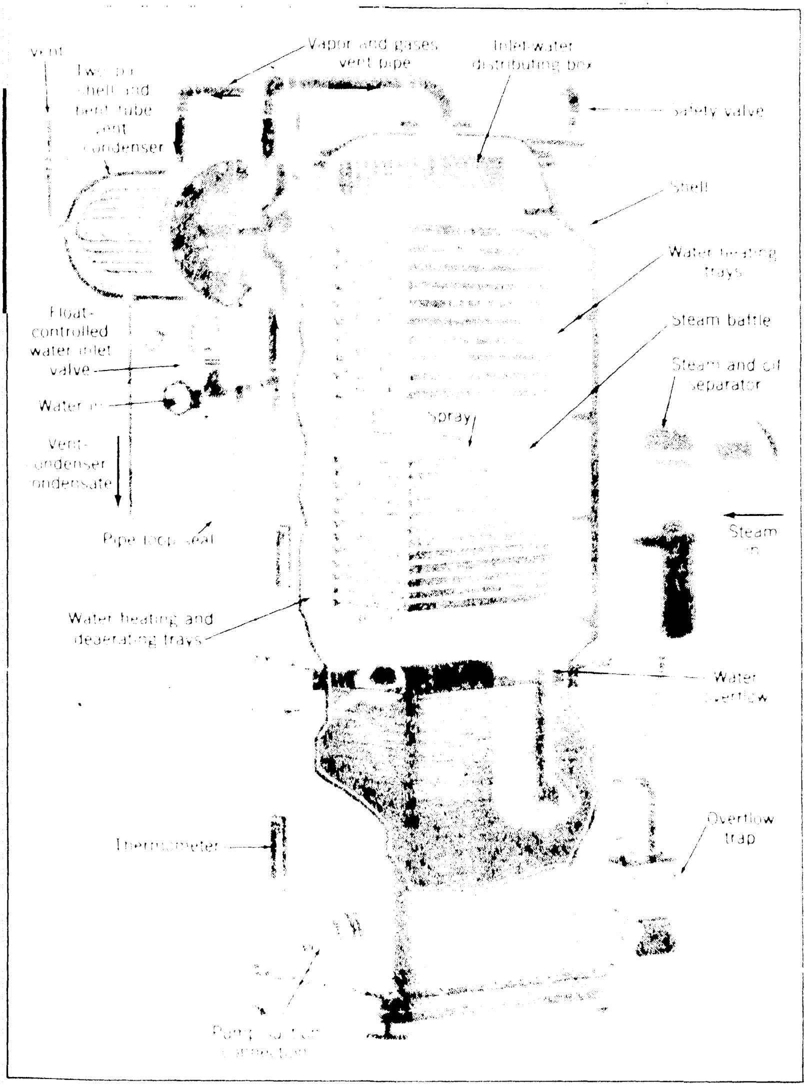


FIG. 91. Swartwout tray-type direct-contact deaerating feedwater heater.

than those having an external unit. The heater, of Fig. 92, has a stainless steel inlet-spray assembly which along with the vent-collecting hood serves as an internal condenser.

The incoming feedwater is sprayed across the opening of the vent-collecting hood to strike the baffles of the inner head. From this loca-

tion the water drops to the bottom of the heating chamber and in passage is heated by the steam of the space which brings the water temperature near to that of the steam. Steam entering at one side of the heater shell passes directly to the deaerating section. Here the steam and the

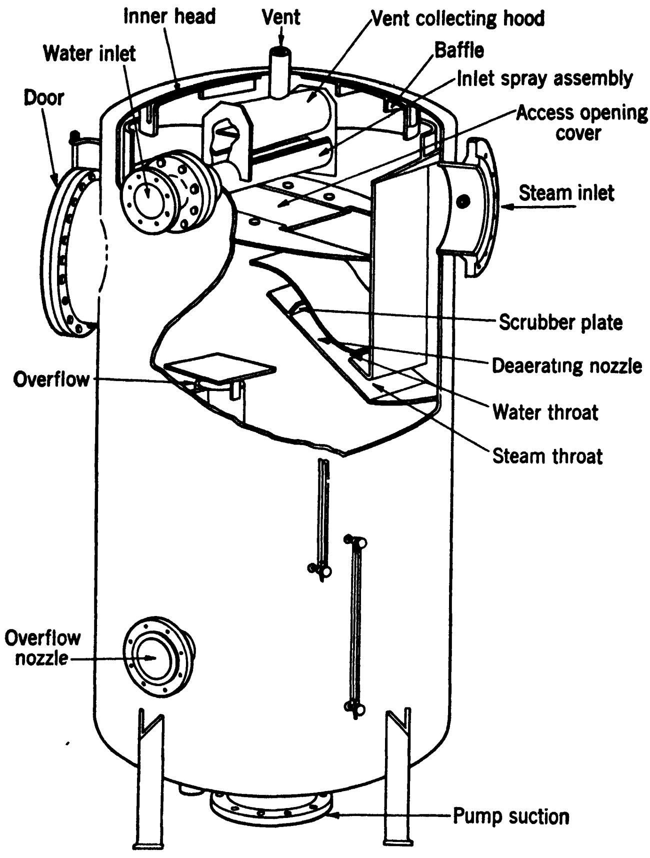


FIG. 92. Elliott direct-contact spray-type deaerating feedwater heater.

water meet after each, respectively, has passed through the throat provided for it. Little water heating is done at this section. However, the water and the steam move at a high velocity through the deaerating nozzle where the water is broken into droplets to be dashed against the scrubber plate and then sprayed with steam into the body of the shell tank. From this point the heated and deaerated water drops to the tank storage section; the uncondensed steam with the released non-condensable gases passes by baffles to the heating section. Here the

steam is condensed in heating the incoming feedwater, and the gases escape through the vent hood and pipe.

132. Heat Transfer of Open or Direct-Contact Heaters. Theoretically the heat given up by the steam should equal the heat absorbed by the water.

$$m_s(h_g - h_{f,2}) = m_w(h_{f,2} - h_{f,1}) \tag{185}$$

where m_s = weight of steam condensed, lb.
h_g = total enthalpy of the steam as condensed, Btu per lb.
$h_{f,1}$ = enthalpy of liquid of inlet water, Btu per lb.
$h_{f,2}$ = enthalpy of the liquid of outlet water, Btu per lb.
m_w = weight of water heated, lb.

In actual operation there are certain inevitable heater losses. Consequently, the value of m_s will always be greater than the amount calculated from the equation for a given quantity of water raised from a temperature of t_1 to t_2 by steam of a given pressure and quality. The heater losses may range from 5 to 10 per cent of the heat of the steam actually used in the heater. The weight of water leaving the heater is the sum of the weight of water supplied and the weight of steam condensed in the water.

133. Closed or Surface Feedwater Heaters. The steam and water are never in contact in the closed or surface type of heater. Consequently, closed heaters may be operated at pressures ranging from high to very low. The essential parts of a closed or surface heater are a shell and the enclosed tubes or coils through which either steam or water is circulated. The tubes may be either straight, spiral, coiled, or U-shaped. The last are commonly spoken of as bent tubes. Heaters having the water inside the tubes or coils are called water-tube heaters; if the steam is confined within the tubes the designation steam-tube heater applies. The water-tube heaters are the most common; the steam and water flow may be either in the same or in opposite directions in either heater, giving either parallel or counterflow conditions. In single-flow heaters the water flows in one direction only; multiflow heaters have the water flowing back and forth in various nests of the tubes. Closed heaters are ordinarily placed in the line between the boiler feed pump and the boilers.

Figure 93 illustrates a straight-tube closed or surface-type feedwater heater having a cylindrical shell. The water makes two passes through the heating-surface bank comprised of small tubes expanded at one end into a fixed head and at the opposite end into a floating head, which moves to relieve strains caused by either the expansion or contraction of the tubes.

Other surface-type heaters have bent water tubes identical in form, but not in size, with those shown in the vent condenser of Fig. 91. Bent-tube construction eliminates the use of the movable or floating head shown in Fig. 93.

Extraction heaters, of the form of Fig. 93, may function as: (1) straight-condensing; (2) desuperheating and condensing; (3) condensing and subcooling; and (4) desuperheating, condensing, and subcooling units. With subcooling operation the temperature of the condensate is reduced

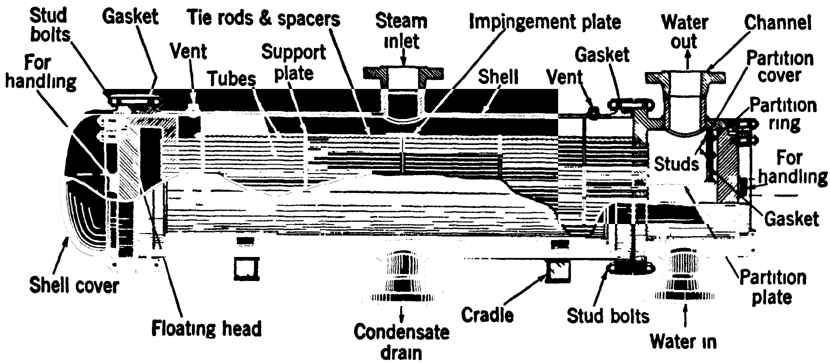


FIG 93 Lummus surface-type extraction feedwater heater.

below that normal for saturated liquid at the steam pressure within the heater. Desuperheating may be accomplished by a tube-bank baffle placed at the steam inlet. Subcooling of the condensate is secured by a shroud enclosure about tubes, through which the water first passes, to prevent steam from coming in contact with them.

134. Heat Transfer in Closed or Surface Heaters. The principles of heat transmission in closed feedwater heaters and surface condensers are essentially alike. If liberal factors are allowed in the design of closed heaters the laws may be taken as the same as those for surface condensers (see Art. 233).

The amount of heat given up in the heater per square foot of heating surface is dependent upon the velocity of the water through the heater and the mean temperature difference between the steam and the water. An increase in either or both of these items will increase the amount of heat taken up by the water. Sufficient time must also be allowed in order that the temperature rise may be a maximum under the conditions of operation. In general, all other conditions being the same, a heater having a large number of passages of small cross-sectional area will transmit more heat than one having a few passages of large cross-sectional area.

In any arrangement of a surface-type heater the required amount of heat-transfer area A, in square feet, may be expressed as

$$A = \frac{m_w(h_2 - h_1)}{U\theta_m} \tag{186}$$

where t_s = temperature of heating steam, F.
t_c = temperature of escaping condensate, F.
t_1 = temperature of entering fluid, F.
t_2 = temperature of leaving fluid, F.
h_1 = enthalpy of entering fluid, Btu per lb.
h_2 = enthalpy of leaving fluid, Btu per lb.
m_w = weight of fluid heated per hour, lb.
U = coefficient of heat transmission, Btu per sq ft per hr per deg F of mean-temperature difference F.
θ_m = logarithmic mean-temperature difference, F.

$$\theta_m = \frac{(t_s - t_1) - (t_c - t_2)}{\log_e \dfrac{(t_s - t_1)}{(t_c - t_2)}} \tag{187}$$

The values of U, for water, may range from 120 to 1500 Btu per sq ft per hr per 1 deg Fahrenheit temperature difference. Average values of U, depending upon the heater type, the steam and water temperature differences, and the velocity of the water flow, range from 200 to 600 Btu.

135. Economizers. Assemblages of water-filled pipes placed in the path of escaping flue gases and used to heat feedwater are known as *economizers*. Their materials of construction are steel and cast iron. Their placements may be in the last flue-gas pass within a boiler setting or in a casing between the boiler setting and the chimney or stack. Internal corrosion is avoided by the use of properly conditioned feedwater and external corrosion by holding the flue-gas temperatures high enough to prevent the formation of an acid.

Economizers increase the resistance to the flow of flue gases and also reduce their temperature. Consequently, natural draft provided by a chimney may be inadequate, and induced draft may be required. The use of economizers is justified when profitable savings of fuel costs can be shown after allowances for all factors of plant economics have been made.

136. Steel Economizers. Many power plants employ economizers constructed of small smooth-exterior steel tubes massed in banks between inlet and outlet headers as shown by Figs. 59 and 94. The tubes with their return and other bends are assembled in position by welding. Provision is made at tube headers for internal inspections and cleanings.

The exterior surfaces are freed from soot and ash accumulations by means of soot blowers.

Another form of steel economizer has straight tubes assembled between headers to form sections which are assembled to give the required water heating surfaces. The marine boiler economizer of Fig. 60 has the

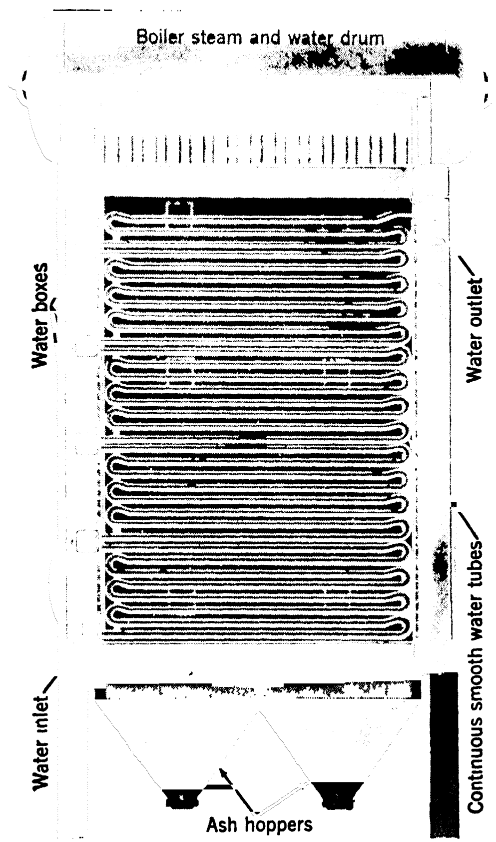


FIG. 94. Erie City continuous steel-tube economizer.

straight portions of continuous-tube loops covered by extended-surface cast-iron rings. These give additional heating surfaces and provide protection to the tube exteriors.

An integral economizer or steaming economizer is illustrated in Fig. 95 in connection with a four-drum Stirling boiler. The economizer tubes extend between the lower header or small drum and the top rear steam-and-water drum of the boiler. The water enters the lower header and

flows upward in the economizer tube bank and then downward through the rear boiler tubes The location of the economizer is in the path of the flue gases as they leave the boiler tubes Soot blowers are used to remove soot and ash accumulations from the economizer tubes.

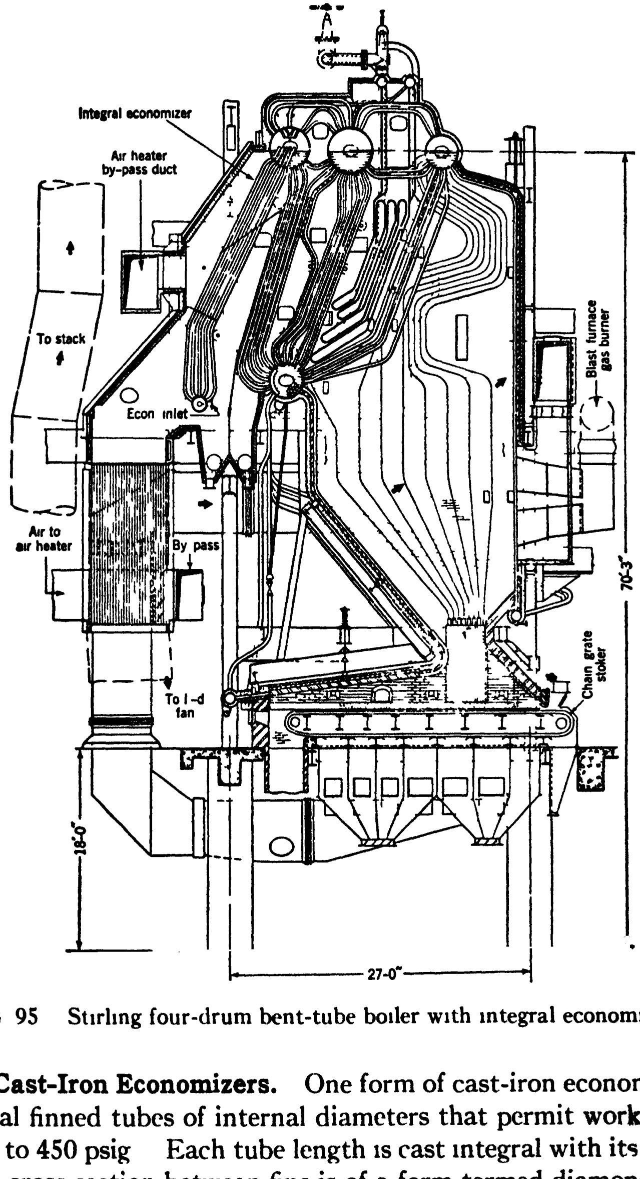


FIG 95 Stirling four-drum bent-tube boiler with integral economizer.

137. Cast-Iron Economizers. One form of cast-iron economizer has horizontal finned tubes of internal diameters that permit working pressures up to 450 psig Each tube length is cast integral with its fins, and the tube cross section between fins is of a form termed diamond shaped

by the maker. Such a design reduces the accumulations of soot and ash deposits on the tube between the fins. The tubes are staggered in the rows and are arranged in tiers not more than six tubes high. Large economizers are assembled in heights of two or three tiers. Flanged return bends connect the tube ends of adjacent horizontal rows. The flow of water is through multiple parallel circuits. Soot and ash deposits are removed by movable soot-blower elements installed at the top of each tier assemblage.

138. Water Impurities. The water of falling raindrops may absorb oxygen, carbon dioxide, nitrogen, dust, and other impurities from the air to become more or less a solvent of minerals of the earth. This pollution may be further augmented by acids from decaying organic materials, industrial wastes, and sewage dumped into lakes and rivers. The materials carried by water may be classified as dissolved and suspended.

Dissolved substances include bicarbonates of calcium, magnesium, and sodium, that is, $Ca(HCO_3)_2$, $Mg(HCO_3)_2$, and $NaHCO_3$; sulfates of calcium, magnesium, and sodium, that is, $CaSO_4$, $MgSO_4$, and Na_2SO_4; nitrates of calcium, and magnesium, that is, $Ca(NO_3)_2$, and $Mg(NO_3)_2$; chlorides of calcium, magnesium, and sodium, that is, $CaCl_2$, $MgCl_2$, and $NaCl$; iron oxide, Fe_2O_3; silica, SiO_2; industrial wastes; and gases, such as oxygen, O_2, and carbon dioxide, CO_2. Suspended materials involve mud, sand, vegetable matter, and industrial wastes.

139. Effects of Water Impurities. Water impurities may be responsible for the following ill effects in boiler and power-plant operation: (1) reduction of the amount of heat transmitted due to scale formation on water heating surfaces; (2) tube and plate failures caused by the reduced amount of heat transmitted through them; (3) embrittlement and corrosion of the boiler steel; (4) bad boiler operation, foaming and priming; (5) cost of cleaning, repairs, inspection, maintenance, and reserve equipment; (6) heat losses due to frequent boiler blowdowns; (7) loss of economy of steam-using equipment caused by dirty steam; and (8) additional cost of condensing equipment for gas-laden steam. Most of the foregoing ills may be attributed to one or more of these items: scale, corrosion, embrittlement, foaming and priming, and occluded gases.

Scale is either a hard or soft deposit on the internal surfaces of a boiler and is composed of mineral materials, or dirt, or both. The effect is to reduce heat transmission through heating surfaces with a reduction of capacity, a loss of operating efficiency, and possible overheating of boiler tubes and shell plates.

Corrosion is a wasting away of boiler parts with a reduction of their

strength. The effects may be due either to electrolytic action, water acidity, water alkalinity, or the presence of oxygen.

Caustic embrittlement and *corrosion fatigue* result from nonuniform corrosion which leads to accelerated cracks in the edges of riveted boiler plates, rivets, and the metal around openings where it has been highly stressed during its fabrication and operation. In welded boilers the chief areas affected are about tube openings. Failure due to caustic embrittlement results from metal cracks which follow grain boundaries and are intercrystalline. Corrosion fatigue, arising from chemical attack and repeated stressing of metal parts, produces cracks which follow lines of stress without regard to the grain boundaries of the metal.

Foaming occurs when alkalies, suspended organic matter, some oils in the presence of soda salts, or waters containing concentrations of soluble sodium salts cause the water in the boiler to fill its steam spaces with a mass of frothy bubbles.

Priming is the discharge of slugs of water with steam; it may be due to the presence of a film of oil on the surface of the boiler water, to the same causes as foaming, or it may occur in boilers having small steam and water disengaging surfaces.

Oxygen, especially at high temperatures, induces pitting and corrosion of all metal parts with which it comes in contact. *Carbon dioxide* alone is likely to produce corrosion or to enter into combination with other materials to form scale-forming compounds. Gases, such as oxygen, nitrogen, carbon dioxide, and others not condensable at the usual condenser pressures and temperatures, that may be carried by the steam impose an additional load upon the air ejectors or air pumps of condensers.

The necessity for either the removal of or the mitigation of feedwater impurities is greatest in plants having large boiler units driven at high steaming rates. Other plants of smaller size and less capacity often operate with a limited amount of water-purifying equipment.

140. Water Hardness. The term "hardness," as applied to water, indicates its scale-forming properties and its destroying power of a standard-soap test solution. Dependent upon the nature of the impurities carried (Art. 138) waters are classified as having either *carbonate* or *noncarbonate* hardness. The amount of any scale-forming materials may be expressed in terms of parts per million, ppm, of equivalent calcium carbonate ($CaCO_3$) carried by the water. If grains per gallon are desired the conversion may be made by dividing the parts per million by 17.1. In natural waters the hardness may range from less than 10 ppm, which is low, to over 1800 ppm which makes a water unusable

for industrial purposes. A complete indication of the desirability of a water for boiler-feed purposes is not given by the contained equivalent scale-forming materials, ppm, because all such impurities are not of equal undesirability.

Waters having carbonate hardness may be softened by either sufficient heating or boiling. In this method of purification CO_2 is liberated, and relatively insoluble precipitates of the carbonates $CaCO_3$ and $MgCO_3$ are formed. In equation form the reaction may be expressed for calcium bicarbonates as $Ca(HCO_3)_2$ plus heat gives $CaCO_3 + CO_2 + H_2O$. The same general equation applies when $Mg(HCO_3)_2$ is substituted in place of $Ca(HCO_3)_2$, and $MgCO_3$ results. The sludge formed may either be discharged with the blow-off water from a boiler or be removed from the feedwater heater. The scales formed by waters having carbonate hardness are softer and more porous than those produced by waters having noncarbonate hardness. Heating alone will not remove calcium and magnesium sulfate from waters having noncarbonate hardness. Sodium salts in solution can be removed only by blowing down a boiler when the concentrations become excessive. Under heat and pressure chlorides and nitrates are prone to break up to form corrosive acids.

141. *p*H Values. Water has the ability to dissolve to some extent everything with which it comes in contact. The rate of corrosion of metals produced by water is dependent upon its temperature, the hydrogen-ion concentration, the amount of dissolved oxygen present, and either the presence or the absence of certain mineral salts. A small part of water, H_2O, is considered to be always partly dissociated so that free hydrogen (H^+) and free hydroxyl (OH^-) ions exist. As the amount of undissociated water is a very large portion of the total amount, the product of the hydrogen ions (acid) and the hydroxyl ions (alkaline) is equal to a constant 10^{-14}. Pure water is neutral so that the numbers of the H and the OH ions are equal, in a given volume, and the concentration of each is expressed as 10^{-7}. As a matter of convenience in indicating the acidity or alkalinity of water the term *p*H is used. This value is the logarithm of the reciprocal of the hydrogen-ion concentration. Thus for pure water the logarithm of the reciprocal of the H^+ ion amount is 7, which is the *p*H value. When the hydrogen-ion concentration is increased its reciprocal becomes smaller; therefore, the logarithm is less, and the *p*H value is lower. Waters having a *p*H value less than 7 are acid and those having a *p*H value greater than 7 are alkaline; that is, *p*H 9 indicates alkalinity. In order to limit corrosion, boiler feedwaters are generally maintained slightly alkaline.

142. Objects of Water Conditioning. The primary objects of water treatment are (1) the removal of suspended and soluble solids, and (2) the removal of gases. These things are necessary to prevent the formation of scale on water-heating surfaces; to provide protection against corrosion of metals in boilers, economizers, and piping; to eliminate carryover deposits of scale on superheater tubes and turbine blades by the prevention of boiler foaming and priming; and to inhibit the possibilities of caustic embrittlement of boiler steels. The foregoing items require that the conditioned water have low limits for hardness factors; satisfactory *p*H values with the elimination of carbon dioxide, oxygen, and other corrosive gases; low contents of dissolved solids, alkalinities, and sludge concentrations; and the maintenance of the proper ratios of sulfates and carbonates in the boiler water.

143. Methods of Water Treatment. Water treatment should be undertaken only after analyses of the water and recommendations for its treatment have been made by an experienced and competent chemist. No single prescribed process of treatment or boiler compound is suitable for all waters. Each case must be considered individually.

Water-treatment processes include removal of debris by stationary or moving screens, filtration, separation of mud and silt in settling basins by subsidence, heating, evaporation or distillation, deaeration, treatment with lime, treatment with soda ash, treatment with both lime and soda ash, lime and barium, trisodium phosphate, coagulants, and zeolite softeners. The processes are not all carried out in the same manner; some of the methods are continuous, others are intermittent. Cold water is treated sometimes but heat is necessary for best results in many methods.

144. Filtration. Mechanical filters have a wide usage in the removal of suspended matter in water, especially that which has been subjected to chemical treatment and which carries particles of precipitates. Pressure filters are built with their steel shells in either a horizontal or a vertical position, depending upon the space available for installation. For either type its best location is on the discharge side of a treated water pump. The flow of water to be filtered is downward through layers of different materials of various sizes. Common filtering materials are sand, magnetite, calcite, and anthracite. Silica in boiler feedwater is very undesirable from the standpoint of the formation of a dense flinty scale. This encrustation of the heating surfaces of steam generators and, when carry-over occurs, those of superheaters and turbine blading is serious. Silica from filter-bed sand is to be avoided; this can be done

by using a filter bed of anthracite, properly sized and uniform in gradings from coarse to fine.

The introduction of coagulants into the water as it passes into the filter aids materially in the maintenance of a rapid rate of filtration. The common coagulants are alum, sulfate of aluminum, or iron sulfate. These chemicals when added to the water react with the alkaline salts produced by bicarbonates or carbonates of lime and magnesia to form a bulky gelatinous precipitate of hydrate of aluminum or iron. This gelatinous hydrate forms a blanket in the filtering material which effectively removes small particles of suspended matter. Automatic feeding devices are provided for the introduction of the coagulant into the water to be filtered.

145. Evaporators. Nearly pure water can be obtained by distillation of the feedwater supply. In plants where surface condensers are used and their condensate is returned to the boilers, the distilled water to be provided is only that necessary to make up the quantities of water disappearing by leakage and unavoidable losses. Distillation of the make-up water supply only is commercially feasible. The use of distilled water almost eliminates the formation of scale and other feedwater difficulties in boiler operation. Even where distilled water is provided, small amounts of scale-forming materials may be carried into the boilers by condenser leakages and by impurities thrown over in the distilling equipment.

Evaporators are used to provide distilled water for boiler feed. Various types of this equipment are built to operate with steam as a heating medium either at high, moderate, or low pressures. The vapor produced by the evaporator may be condensed in the feedwater heaters by the incoming feedwater or in separate evaporator condensers using the feedwater to be heated as the cooling water. The most common type of evaporator has steam-filled tubes submerged in the water to be evaporated.

Evaporators are also further designated as being single- or multiple-effect units, depending upon the number of evaporating stages through which the water is passed. Thus a single-effect unit is one in which the vaporization is completed in a single coil or stage. A double effect is secured by condensing the vapor from a single-effect unit in the coils of a second evaporator. The second evaporator operates at a lower temperature, and from it the vapor passes to the feedwater heater or the evaporator condenser. Multiple-effect evaporators utilize more of the heat of the steam supplied to them, but they do not give any greater evaporating capacity per square foot of heating surface than single-effect evaporators. The low-pressure evaporators usually exhaust to

open types of feedwater heaters at about atmospheric pressure; the high-pressure types of evaporators exhaust to special evaporator condensers.

The evaporator shown in Fig. 96 embodies a shell-and-tube arrangement which is a common form of construction. The bundle of straight

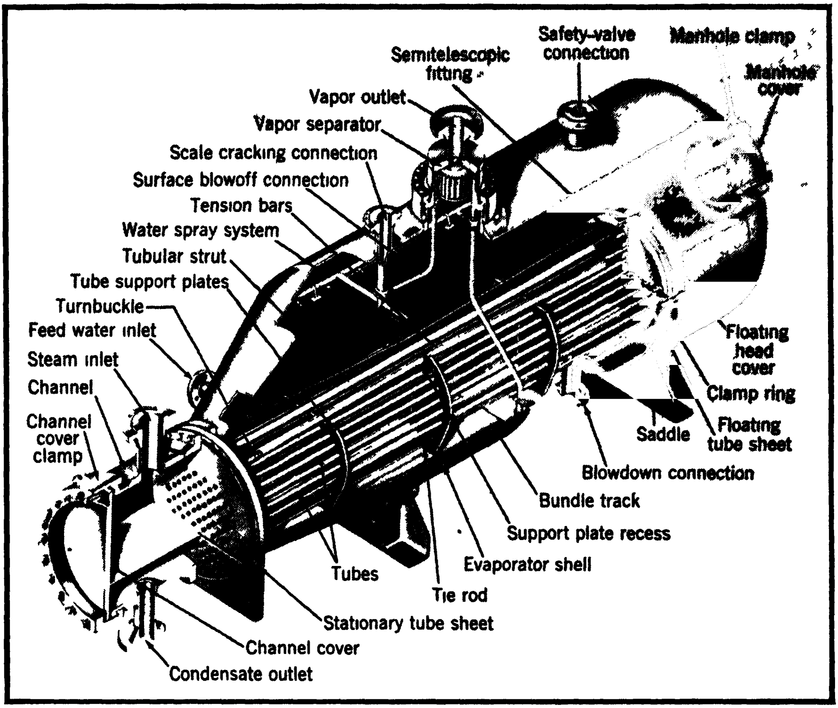


FIG 96 American Locomotive Company flexible-tube evaporator.

steam-filled tubes is submerged in the water to be evaporated, which is maintained with its level at midheight of the evaporator shell. The feedwater inlet is at a level just below the top of the tube bank. Water vapor and any separated gases rise to fill the upper portion of the evaporator shell. As the vapor leaves the evaporator it passes through a separator which returns water particles to the lower part of the shell. Usually some form of surface condenser serves to reduce the vapor to liquid and to vent any released gases. The unit may be installed for either single-effect vacuum operation or for single-effect low-pressure conditions. Two of the evaporators can be arranged to give double-effect pressure operation and three to give triple effect.

The removable tube bundle is assembled when cool with a fixed tube

sheet, a floating tube sheet, two pairs of tension (flexing) bars with the members of each pair initially deflected toward each other, intermediate and divided tube-support plates equally spaced, and a pair of tubular struts. Under conditions of assembly the tubes are straight but when they are filled with steam their expansions under the restraints placed upon them are such that one half of them bow horizontally to the right and the other half to the left. Movements of the tubes function in the removal of scale deposits. Certain scale formations on the tubes may be removed by withdrawing the water from the evaporator and spraying cold water upon the hot tubes. The contraction of the tubes when suddenly cooled loosens the scale. When this process is ineffective for the deposits involved, the tube bundle may be withdrawn from the shell and the encrusting materials removed by some manual operation.

146. Deaerators. Mechanical devices for liberating feedwater gases (air, oxygen, carbon dioxide, or other gases) are known as deaerators. In their operation the feedwater is broken up into fine droplets and heated to produce vapor within the deaerator, and, in the process, air, carbon dioxide, and other gases are separated from the vapor as it condenses. Steam is the heating medium in deaerators at pressures ranging from high to less than atmospheric.

A feedwater heater of the open or direct-contact type inherently serves as a deaerator provided that the water is heated to a temperature high enough to drive off the gases held within the water. The gases escape through the vent of the heater. As a matter of fact most modern deaerators are direct-contact type of feedwater heaters similar in principles to those of Figs. 91 and 92. These units can be constructed to give very low limits of oxygen and other gas contents in the water handled. The distinction between a deaerating direct-contact feedwater heater and a deaerator, in such cases, is the low limit for oxygen content of the water to which the deaerator can work.

Power plant equipment for feedwater deaeration embodies the tray type and the atomizing units. Very corrosive waters are sometimes deaerated by subjecting them to low absolute pressures without heat. For hot-water service in buildings deaeration may be effected by heating without having the steam and the water in contact.

147. Water Treatment with Chemicals. The use of various compounds for treatment of water after it is within a boiler is applicable only under very limited conditions. No single method of chemical treatment is applicable to all waters. Chemical treatment of feedwater involves both cold and hot processes. Generally the heating of the water hastens the precipitation of dissolved materials, gives softer water, and requires smaller amounts of chemicals and smaller tank

equipment than when the water is unheated. A few of the more common chemical reactions of water softening follow.

Lime in a hydrated form $Ca(OH)_2$ is suitable for the treatment of carbonate hardness. The carbonic acid CO_2 in the water, either in a free state or in bicarbonate combination, is taken up and the relatively insoluble precipitates of calcium carbonate and magnesium hydroxide are formed. Respectively, the reactions are

$$CO_2 + Ca(OH)_2 \rightarrow CaCO_3 + H_2O$$

$$Ca(HCO_3)_2 + Ca(OH)_2 \rightarrow 2CaCO_3 + 2H_2O$$

$$Mg(HCO_3)_2 + 2Ca(OH)_2 \rightarrow Mg(OH)_2 + 2CaCO_3 + 2H_2O$$

$$2NaHCO_3 + Ca(OH)_2 \rightarrow CaCO_3 + Na_2CO_3 + 2H_2O$$

The process is best carried on in large tanks from which the softened water may be decanted and the sludge removed from the treating plant.

The *soda* process is suitable for water containing noncarbonate hardness; soda ash or sodium carbonate Na_2CO_3 being commonly used. As a result of soda treatment decomposition of the sulfates occurs together with the precipitation of insoluble calcium carbonates and soluble sodium sulfate. Reactions in water treatment with soda ash are

$$CaSO_4 + Na_2CO_3 \rightarrow CaCO_3 + Na_2SO_4$$

$$CaCl_2 + Na_2CO_3 \rightarrow CaCO_3 + 2NaCl$$

The *lime and soda* process is a common method where both carbonate and noncarbonate hardness exists. The lime takes up the CO_2 not affected by the soda used in the treatment of noncarbonate hardness. The principal reactions with the combination treatment are

$$MgSO_4 + Ca(OH)_2 + Na_2CO_3 \rightarrow Mg(OH)_2 + CaCO_3 + Na_2SO_4$$

$$MgCl_2 + Ca(OH)_2 + Na_2CO_3 \rightarrow Mg(OH)_2 + CaCO_3 + 2NaCl$$

Figure 97 illustrates a hot-process treating plant suitable for the use of either lime or soda ash or both in combination. The chemical solution is drawn from the storage tank, and properly proportioned amounts are mixed with the raw water as it passes the softener spray head. The water is discharged through the spray head into low-pressure steam which fills the upper portion of the shell and which heats the water by direct contact with it. The water moves downward, and the precipitated materials are deposited at the bottom of the softener body where they may be withdrawn through the sludge outlet. The purified water is removed above the level at which the sludge collects and is normally passed through a filter.

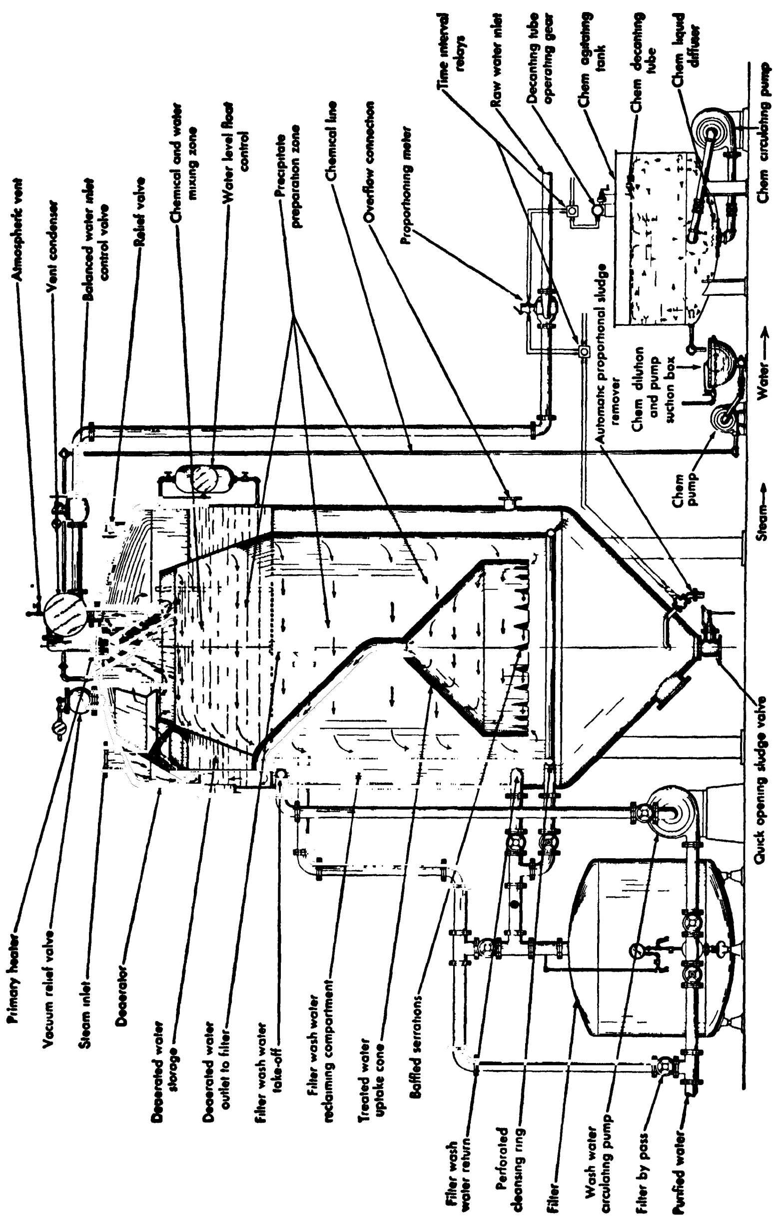


FIG 97 Worthington hot lime soda (deaerating type) water softener.

148. Zeolite Treatment. Either natural or artificial hydrated silicates of sodium and aluminum, having in general the formula $Na_2O \cdot Al_2O_3 \cdot SiO_2$, are known as sodium zeolites Na_2Z. These materials have the property of removing calcium and magnesium from water passed through them as their bases are interchangeable. Thus in the softening process sodium of the zeolite goes into solution as either sodium carbonate, sodium sulfate, or sodium chloride as calcium and magnesium are absorbed from the water by the zeolite material. The base exchanges are

$$Na_2Z + \left.\begin{matrix} Ca \\ \text{and/or} \\ Mg \end{matrix}\right\} \left\{\begin{matrix} (HCO_3)_2 \\ SO_4 \\ Cl_2 \end{matrix}\right. = \left.\begin{matrix} Ca \\ \text{and/or} \\ Mg \end{matrix}\right\} Z + \left\{\begin{matrix} 2NaHCO_3 \\ Na_2SO_4 \\ 2NaCl \end{matrix}\right.$$

and no precipitates are formed. Zeolite treatment produces waters with very low calcium and magnesium contents. When the zeolite material becomes inactive it is regenerated by washing it with salt (NaCl) brine to restore its sodium by exchange.

$$CaZ + 2NaCl \rightarrow Na_2Z + CaCl_2$$

$$MgZ + 2NaCl \rightarrow Na_2Z + MgCl_2$$

Natural zeolites (green sand) are suitable for use with cool water free from acids. These materials find usages for certain purposes but in many cases they have been supplanted by special artificial resinous products (also termed zeolites) developed for the function of water softening. The newer agents are able to withstand high temperatures, acids, and alkalies, and under some conditions they can effect changes of both anions and cations of impurities dissolved in water. In any event, water passing through a zeolite bed should be free from debris, mud, silt, and finely divided precipitates which coat and clog particles of the softening materials to render them less effective.

149. Sodium Zeolite Water Softening. A sectional view of a downflow zeolite water softener is illustrated by Fig. 98. The water enters through a distributing manifold at the top of the softener unit. The flow is downward through the zeolite bed and graded sizes of quartz which serve as a supporting medium for the ion-exchange material. This zeolite bed serves to remove suspended matter carried by the water, although it is not intended to serve as a filtering agent. Suspended matter should be removed from water prior to its entering a zeolite softener. The softened water is discharged from the assembled parts through the lower collecting manifold.

During its regeneration the zeolite bed is washed free of sediment and is regraded by the discharge of backwash water through it. Then

sodium chloride (NaCl) solution is drawn from the brine tank by means of a hydraulic ejector or brine pump and is distributed over the zeolite material to flow through it to produce a base exchange. The final step in the regeneration is the rinsing which washes out the excess salt, together with calcium and magnesium chlorides which have been formed The flows of water through the softener are controlled by a single multiported valve.

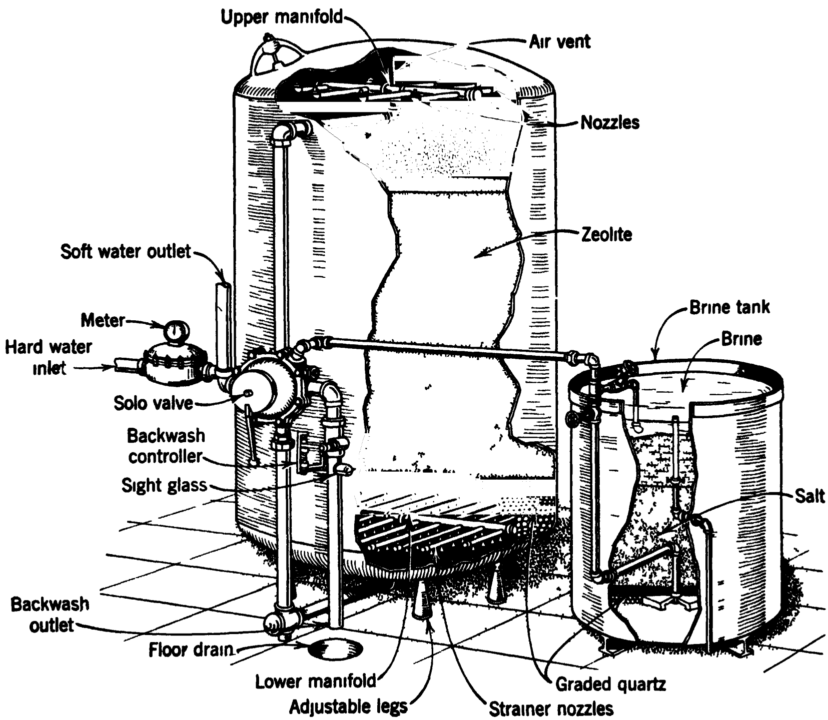


FIG 98 Elgin downflow zeolite water softener.

150. Hydrogen Zeolite Softeners. Waters treated solely by sodium zeolite carry a considerable amount of dissolved sodium salts which may produce foaming and priming in a boiler. In order to overcome this difficulty, hydrogen zeolites may be used These artificial resins, especially compounded for specific purposes, are regenerated with sulfuric acid, H_2SO_4. The reactions of H_2Z with $Ca(HCO_3)$, $Mg(HCO_3)_2$, and $2NaHCO_3$ produce CaZ, MgZ, and Na_2Z together with $2H_2O$ and $2CO_2$. When H_2Z comes in contact with calcium magnesium, and sodium sulfates, respectively, sulfuric acid (H_2SO_4) is produced with CaZ,

MgZ, or Na_2Z as the case may be. When the chlorides of calcium, magnesium, and sodium are handled, 2HCl appears along with CaZ, MgZ, or Na_2Z. The regeneration with H_2SO_4 produces H_2Z, $CaSO_4$, $MgSO_4$, and Na_2SO_4.

151. Combinations of Sodium and Hydrogen Zeolite Systems. In the hydrogen zeolite method of water softening the treated water, with its total dissolved solids, has an acid character. To effect a control of

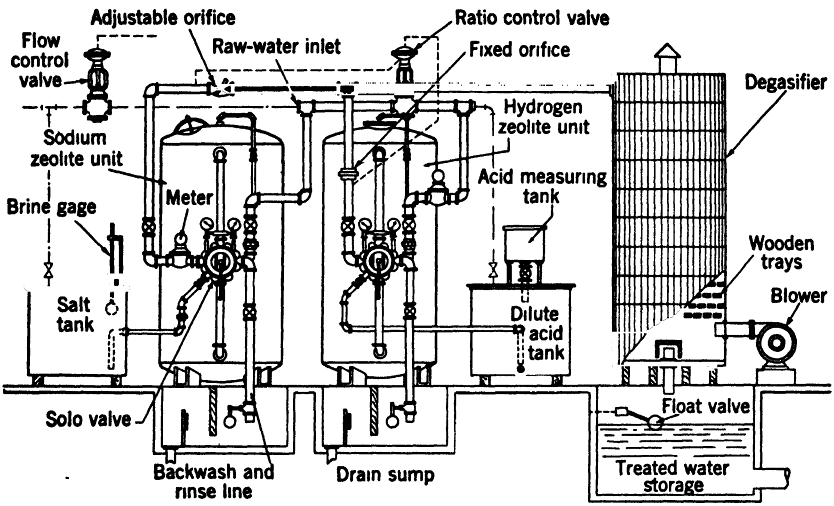


FIG. 99. Allis-Chalmers sodium zeolite and hydrogen zeolite softener for the production of demineralized water.

water which involves hardness, alkalinity, total dissolved solids, and CO_2 which is corrosive, hydrogen zeolite softeners are used in combination with sodium zeolite units. The arrangement of apparatus shown by Fig. 99 indicates one method of control. The essential steps in the operation are ion exchange, neutralization, and degasification. The equipment of Fig. 99 functions to mix together proportionate parts of the discharges from the softeners. As shown in Fig. 99 water from the sodium unit is blended with effluent from the hydrogen zeolite softener. Another manufacturer supplies similar equipment to send a portion of the sodium-softener water through the hydrogen unit and then to blend water from the latter with water which has come directly from the sodium zeolite unit. For boiler-feed purposes the water should always be slightly alkaline. In the degasifier the water is broken into droplets passing over wooden trays and is exposed to air supplied by a blower. This process is to reduce any CO to CO_2 and to eliminate the latter from

the water processed. Controls are necessary to prevent water from the hydrogen unit passing into the system if flow through the sodium unit ceases. The proportioning of water is accomplished by a valve operated by a pressure differential created by an adjustable valve in the sodium-unit output line and an orifice in the hydrogen-unit discharge piping. In some cases satisfactory results may be obtained by mixing hydrogen zeolite water with small amounts of raw water, or in other cases the acid water is neutralized with chemicals before degasification.

152. Water Deionization. In this process a single tank may hold a mixture of cation and anion resins. The cation agent is a sulfonated polystyrene resinous hydrogen-ion exchanger. For the anion substitution a strongly basic amine-type exchanger resin is suitable. The cation exchange resin removes positively charged ions such as calcium, magnesium, iron, sodium, and potassium and replaces them with hydrogen, H^+, ions. The anion resin eliminates negative ions such as sulfates, chlorides, nitrates, carbonates, carbon dioxide, and silica from the solution and substitutes hydroxyl ions, OH^-, for them. Water is formed as the hydrogen and the hydroxyl ions neutralize each other. The combined operations eliminate minerals, silica, and carbon dioxide to give approximately neutral water. By careful manipulation of the backwash operation the intermixed cation and anion resins are separated into two zones. A dilute solution of sodium hydroxide is passed through the anion resin to regenerate it. This operation is followed by a short soft-water rinse. Dilute sulfuric acid is required to render the cation resin active. Air and water admitted at the bottom of the deionizer shell intermix the resins, and the unit is again ready for operation.

153. Hot Zeolite Water Treatment. Zeolite softeners capable of handling heated water may be attached to the discharge line of a hot-process water softener. A pressure filter may be a desirable piece of equipment in the connecting piping. In the combination unit two chemicals are required: lime for the hot deaerating softener and salt (NaCl) for reconditioning the zeolite. The combination of units produces water low in dissolved minerals, without excess alkalinity, with small concentrations of oxygen, little silica, low alkalinity, high *p*H, small amounts of CO_2 in the steam produced, and diminished precipitation of phosphates in the steam-generator equipment served.

154. Inhibition of Caustic Embrittlement. The design and construction of welded boiler drums reduce the number of focal points in seams and rivet holes as locations of caustic embrittlement. Care should be taken to prevent overstressed metal about and in the ends of tubes expanded in tube sheet openings by a rolling process. Proper water conditioning is an aid in reducing the agents of localized attack such as silica and sodium hydroxide which may result from sodium salts in

solution. The addition of a sufficient quantity of sodium nitrate offers protection when free caustic is present. Organic materials such as quebracho, tannin, and waste sulfite are effective in some cases.

PROBLEMS

1. Changes are made in the operation of a boiler plant so that the temperature of the feedwater is 200 F instead of 140 F. The plant uses coal which has a heating value of 11,000 Btu per lb as fired with an over-all efficiency of 70 per cent for either condition of operation. What additional weight of water may be evaporated per 2000 lb of coal fired, if the steam pressure is 200 psia and the quality of the steam is 0.97? 860 lb, *Ans.*

2. A boiler plant operates normally with a pressure of 145 psia and steam of 0.98 quality. What percentage of increase of fuel consumption may be expected if the efficiency of operation is unchanged when the temperature of the feedwater is reduced from 175 to 55 F?

3. Steam at 18 psia and 0.90 quality is used in a direct-contact heater to raise the temperature of 250,000 lb of incoming water from 55 to 200 F during a period of 1 hr. Find the weight of steam required if the heater losses are 5 per cent of the heat obtained from the steam condensed.

4. Thirty-five thousand pounds of steam are condensed in a direct-contact heater each hour. The steam has a pressure of 15 psia and a quality of 0.96. The heater losses amount to 10 per cent of the heat obtained from the steam Find the pounds per hour of water which may be heated from 60 to 180 F.

253,500 lb, *Ans.*

5. A surface-type heater has 525 sq ft of heating surface, and operates with a coefficient of heat transmission, U, of 360 Btu per (hr) (sq ft) (deg F). Steam is supplied at 60 psia and 0.99 quality. How many pounds of water may be heated per hour from 60 to 260 F? 96,200 lb per hr, *Ans.*

6. Forty-thousand gallons of water, measured at its entering temperature of 70 F, are heated to 280 F by steam supplied to a surface-type heater at 80 psia and 0.95 quality. The period of heating time is 1 hr, and the coefficient U may be taken as 370 Btu per (hr) (sq ft) (deg F). Find the required amount of heating surface and the amount of steam per hour, including 10 per cent losses.

7. An economizer operates with heat losses amounting to 7 per cent of the heat abstracted from the dry flue gases passing through it. The conditions of operation are: temperatures of flue gases at the economizer, entering 550 F, leaving 400 F; mean specific heat of flue gases 0.26; temperature of water entering 190 F, leaving 400 F; and weight of flue gases 235,000 lb per hr. Find the weight of water that may be heated per hour.

8. A steam generator produces 825,000 lb per hr of steam. The rise of feedwater temperature in surface-type heaters is from 80 to 285 F and in the economizer from 285 to 450 F. Steam is produced in the boiler at 750 psia and 0.98 quality and is discharged from its superheater at 725 psia and a temperature of 660 F. Find the change of enthalpy in each piece of equipment, and express it as a percentage of the total change of enthalpy in all the pieces of equipment.

9. A direct-contact heater functions to raise the temperature of 140,000 lb of water per hr from 80 to 213 F when supplied with dry steam at 15 psia. The heater thermal losses amount to 8 per cent of the heat energy added to the water. Further heating occurs in a surface-type unit which receives the water at 210 F and increases its temperature to that of saturated steam at 60 psia. Find the required heater surface, in square feet, if U is 400 Btu per (hr) (sq ft) (deg F).

CHAPTER

9

Draft Apparatus
Chimneys and Fans

155. Draft. The pressure difference known as *draft* is usually measured in terms of inches of water gage (in. WG). Its measurements are made either above or below atmospheric pressure. A draft gage, with an inclined scale of small range, is shown in Fig. 100. Draft is necessary for the operation of a boiler furnace in order to supply it with air for the combustion of fuel and to cause the flow of the products of

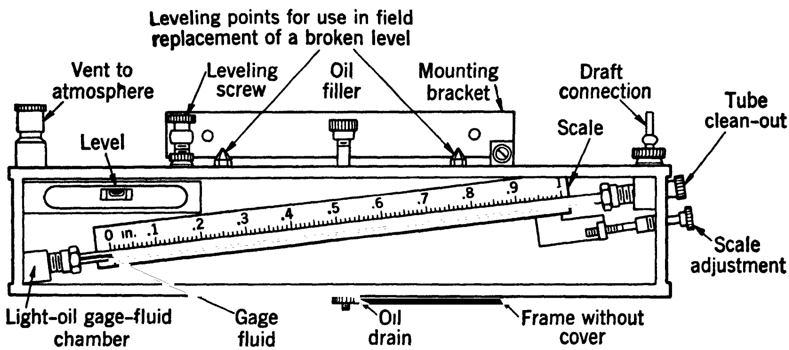


FIG. 100. Ellison inclined draft gage.

combustion from either a chimney or a stack. Draft is designated as either *natural* or *mechanical.* Natural draft is produced thermally; mechanical draft results from the use of either a steam jet or a fan.

Resistances to the flow of air and flue gases, which make draft necessary, are: those of ducts, stacks, chimneys, fuel beds, boiler gas passages, baffles, type of boiler and its setting, dampers, air preheaters, economizers, accumulations of soot and ashes in gas passages, and the required rate of combustion.

156. Natural Draft. The pressure difference termed natural draft is produced either by the action of a stack or a chimney. Its amount

is dependent upon the height of the chimney top above the level of the furnace grates and upon the average temperature difference between the flue gases within the chimney and the outside air. Weather variations, boiler operating conditions, and the chimney height have a marked effect upon the amount of draft that may be secured with a particular chimney.

Natural draft does not require the expenditure of mechanical power in its production and is suitable for small plants where either the load conditions are not extremely variable or where the boilers are not required to develop great capacities quickly.

157. Chimneys and Stacks. Either of these structures is used with all draft systems for conveying smoke or flue gases to an elevation above that of surrounding objects and buildings. Structures for the disposal of flue gases or for the production of natural draft, if constructed of brick, concrete, or radial brick, are known as chimneys; those built of sheet steel or steel plates are designated as stacks. Stacks built of light sheet metal require supporting guys; those built of heavy steel plates may be self-supporting. A typical radial-brick chimney is shown in Fig. 101.

Brick and tile chimneys are likely to have leakages as the result of either poor bonding of the materials or cracks due to settlement of the structure. Concrete chimneys are also subject to cracking. Air leakages into a chimney producing natural draft reduce the amount of the draft available at the smoke connection. Steel stacks, unless

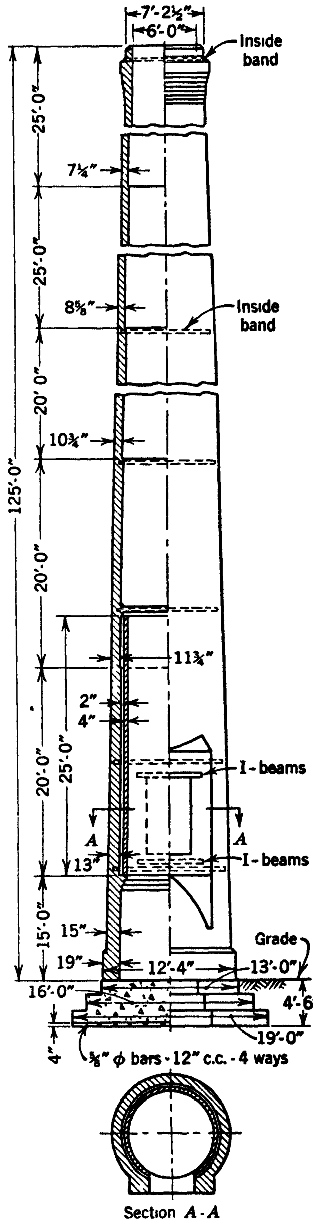


FIG. 101. Consolidated Chimney Company radial-brick chimney.

kept carefully painted, are subject to the corrosive action of the air and the weather. Unlined stacks are corroded by the constituents of the flue gases so that the life of an unlined stack is much shorter than that of a chimney built of bricks, radial bricks, or concrete.

158. Chimney Height and Area. The head available for producing gas flow and overcoming frictional resistances, measured in pounds per square foot, is equal to the chimney height, h feet above the grate level, times the difference between the density of the outside air d_a and the average density of the flue gases within the chimney d_g. This pressure difference is $h(d_a - d_g)$. The pressure equivalent of 1 in. of water in pounds per square foot is $D/12$ when D is the weight of 1 cu ft of water at the draft-gage fluid temperature. The theoretical draft in inches of water is $h_w = 12h(d_a - d_g) \div D$. The density of the outside air and the average density of the flue gases may be computed by means of the gas-law equation $PV = mRT$. The density of either the air or the gas is equal to m when V equals 1 cu ft. The value of R may be taken as 53.34 ft-lb per deg F for both air and flue gases. When the barometric pressure B, in inches of mercury, is known, $P/R = (144 \times 0.491B)/53.34 = 1.326B$, and d_a and d_g are equal to $1.326B/T_a$ and $1.326B/T_g$ when T_a and T_g are the absolute temperatures of the air and the flue gases, respectively. Hence

$$h_w = 15.91Bh\left(\frac{1}{T_a} - \frac{1}{T_g}\right) \div D \tag{188}$$

For a required theoretical amount of draft h_w in inches of water, the chimney height in feet must be

$$h = (h_wD) \div 15.91B\left(\frac{1}{T_a} - \frac{1}{T_g}\right) \tag{189}$$

The theoretical amount of draft is seldom obtained with a chimney, and the actual draft may be 0.8 of the theoretical draft possible. The theoretical draft h_c expressed in feet head of flue gases is $h(d_a - d_g) \div d_g$, or $1.326Bh\left(\frac{1}{T_a} - \frac{1}{T_g}\right)\frac{T_g}{1.326B}$, which simplifies to $h_c = h\left(\frac{T_g}{T_a} - 1\right)$.

The theoretical velocity of gas flow is $V = \sqrt{2gh_c}$ fps. In terms of chimney height and temperatures

$$V = 8.02\sqrt{h\left(\frac{T_g}{T_a} - 1\right)} \tag{190}$$

Under actual conditions of operation the flue-gas velocities may be only 30 to 50 per cent of the theoretical velocities, owing to roughness of the

interior surfaces of the chimney. The cross-sectional area of the chimney, in square feet, is

$$A = Q \div KV \qquad (191)$$

where Q = volume of the gases handled, in cfs. K = coefficient of velocity, 0.3 to 0.5. V = theoretical gas velocity in fps.

Example. A steam boiler plant consumes 20,000 lb of coal per hr and produces 20 lb of dry flue gases per pound of coal fired. The outside-air temperature is 90 F, the average temperature of the flue gases entering the chimney is 650 F, and the average temperature of the flue gases in the chimney is 500 F. The gage-fluid density is 62.11 lb per cu ft, and a theoretical draft of 0.9 in. of water at the chimney base is needed when the barometric pressure is 29.92 in. of mercury. Find the chimney dimensions that are required.

Solution. The velocity coefficient is taken as 0.4 and the density of the gases as that of dry air at 650 F, or 0.0359 lb per cu ft.

$$h = \frac{0.9 \times 62.11}{15.91 \times 29.92(\frac{1}{550} - \frac{1}{960})} = 151 + \text{ft}$$

$$V = 8.02\sqrt{151(\tfrac{960}{550} - 1)} = 85 \text{ fps}$$

$$Q = (20{,}000 \times 20) \div (60 \times 60 \times 0.0359) = 3095 \text{ cfs}$$

$$A = 3095 \div (85 \times 0.4) = 91 \text{ sq ft}$$

The diameter of the chimney should be $\sqrt{(4 \times 91) \div 3.1416} = 10.76$ ft.

The chief weakness in the foregoing method of design is in the assumption of the coefficient of velocity and the ratio of the actual draft to the draft theoretically possible. Most chimney equations are empirical and embody similar assumptions.

159. Mechanical Draft. Draft produced by the action of steam or air jets or centrifugal fans and blowers is termed *mechanical draft.* It is required where positive and certain draft must be maintained irrespective of weather or boiler operating conditions and where the natural draft provided by stacks or chimneys is deficient. Auxiliary boiler equipment, such as economizers and air preheaters, reduces the flue-gas temperatures. In such cases a chimney necessary to produce the required amount of draft may be excessive in height and cost so that mechanical draft is necessary. In general, mechanical draft is desirable when boilers are to be operated at capacities greatly in excess of their rating or when they must quickly supply sudden demands for steam.

Boilers provided with mechanically produced draft do not require stacks or chimneys so high or so expensive as those necessary for natural draft. Sometimes where mechanical draft is used chimneys of considerable height are built to prevent the smoke and flue gases, together with the cinders and fine ashes that may be discharged, from being

objectionable to adjacent property owners. The initial cost, maintenance, depreciation, and operating costs of mechanical-draft equipment must be balanced against similar costs of stacks or chimneys necessary to produce the same amount of draft, when a selection is to be made between the two systems.

160. Mechanical Draft Systems. Two systems are common: *forced* draft and *induced* draft. Either system may be used alone or in combination as illustrated by Fig. 57.

Forced draft is secured by blowing air into a closed boiler room, into chambers or air boxes under stokers or grates, or through pulverized-fuel burners. Air is supplied under pressure and thus forces its way through the fuel bed or burner into the furnace. The general practice in forced draft is to remove the products of combustion from the boiler setting by natural or induced draft or a combination of the two.

Induced draft is procured by a jet blower or a centrifugal fan located in the smoke connection between the boilers and the stack or at the base of the stack. When an economizer is installed the induced-draft equipment is placed between the outlet end of the economizer and the stack. The action of induced-draft apparatus is to reduce the pressure of the gases within the boiler setting below that of the atmosphere. The gases are discharged into the stack under a positive pressure.

Induced-draft appliances may produce a pressure within the furnace well below that of the atmosphere so that the outside air pressure is sufficient to force the requisite amount of air through the fuel bed. With underfeed stokers, forced draft is required for the air supply. When a combination of forced and induced draft is used with practically atmospheric pressure in the furnace above the fire, the term *balanced draft* is applied. Article 106 describes boiler units which have furnaces operated with the products of combustion under pressures up to 30 psig.

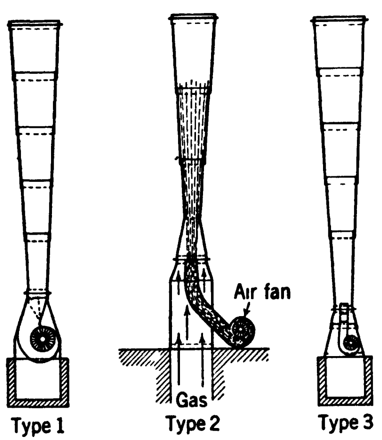


FIG. 102. Prat-Daniel combined fan stacks.

161. Combined Fan Stacks. Three types of these stacks are illustrated in Fig. 102. The type 1 stack is suitable for most induced-draft requirements but is usually selected for installations where more than 1½ in. of water differential pressure must be maintained. As a

rule the type 3 unit finds usage where the necessary drafts are less than $1\frac{1}{2}$ in. of water. Either type of stack may be operated to give either full induced draft or natural draft without the fan in operation. Where corrosive or very hot gases are handled, type 2 stacks, which have the fan located externally, may be used; in this case the fan blows a jet of cool air through the ejector nozzle. Type 1 stacks are built with single and twin fans.

The stack sides form an angle of about 7 deg with the vertical, and an ejector action is secured as the gases pass at high velocities through the constricted section. The reduction of the velocity of the gases is gradual in the flared part of the structure, and a considerable amount of resistance to gas flow is eliminated. A saving in the amount of power required to handle the gases is effected. The stacks are comparatively light in weight and usually have very short smoke-pipe connections to the boilers.

162. Fans. When gases are to be moved against small pressure differentials (0 to 15 in. WG) fans are used. They have wide applications in power plants, drying systems, heating, ventilation, air conditioning, conveying, and cooling.

Essentially a fan consists of a rotating wheel or impeller surrounded by a stationary member designated as a housing. The National Association of Fan Manufacturers, Inc., names and defines types of fans, as indicated in Fig. 103. Energy is transmitted to the gas handled by the power-driven rotating wheel, and a pressure difference is created to produce gas flow. The term fan is applied to those gas-handling pieces of equipment which do not increase the density of the gas by more than 7 per cent as it passes through the device. Whatever their type of construction, fans may be identified as either blowers or exhausters. For the most part blowers discharge gases against a pressure at their outlet. Exhausters remove gases from a space by suction and discharge it under slight pressure at the housing outlet. Certain types of fans working within a given pressure range may serve both as an exhauster and a blower. The fan types of Fig. 103 may be placed in the classifications of axial-flow (propeller, tubeaxial, and vaneaxial) and centrifugal or radial-flow units.

163. Axial-Flow Fans. In these units the flow of the gaseous fluid is essentially parallel to the longitudinal axis of the shaft upon which a propeller or disk type of wheel is mounted. Each of the axial-flow fans of Fig. 103 has its rotating member mounted upon the shaft of the driving motor. The fan impeller may be provided with a separate shaft and suitable bearings so that the driving motor may be placed outside the gas stream. The motor mounting may be attached to the

fan-mounting ring or housing, and the transmission of power from the motor shaft to the fan shaft may be made by means of V belts and pulleys. The blades of axial-flow fan impellers vary as to numbers, shape, and adjustability, their angles with respect to the wheel shaft, their materials and methods of construction, and the ratios of the hub diameters to the wheel diameters.

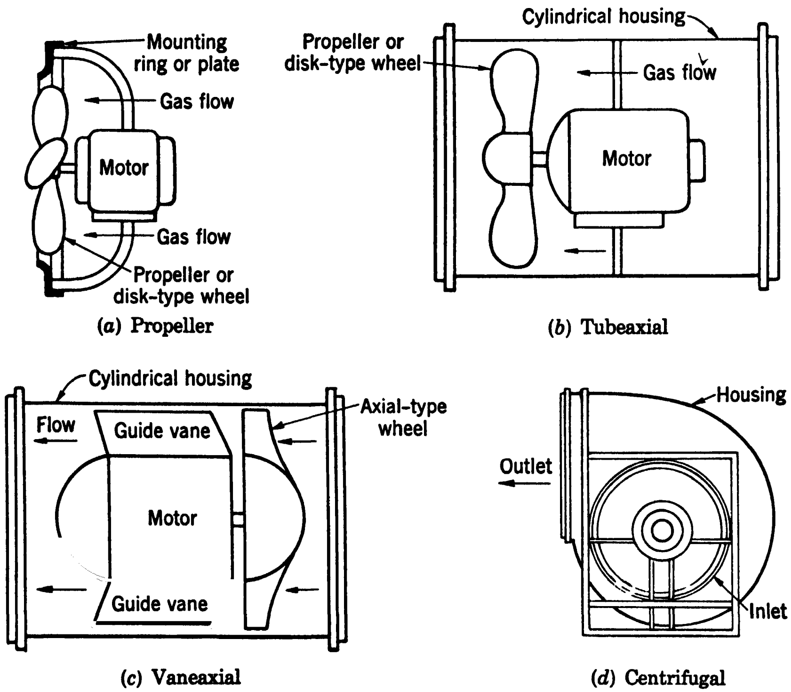


FIG. 103. Fan types. (Used by permission of National Fan Manufacturers Association, Inc.)

The rotating wheels of *propeller fans*, Fig. 103(*a*), are centered within a mounting ring or plate which surrounds their outer periphery. The design of a mounting ring is important in that it functions to prevent gas discharged from the blade tips from being drawn backward into the wheel at that location with a loss of efficiency of operation. Blades of air-foil shapes find usage in propeller fans, and such members may have an adjustable pitch angle with the wheel shaft. Propeller and disk fans are mostly employed to move air against small resistances, such as ventilating units or those used to discharge air into a space beneath a grate upon which a solid fuel is burned.

Both *tubeaxial*, Fig. 103(*b*) and *vaneaxial* Fig. 103(*c*) fans have tubular casings, helical blades mounted upon large hubs, and the ability to work against resistances of 9 to 10 in. WG. Tubeaxial fans are not fitted with guide vanes of any form. Vaneaxial fans have fixed directional vanes, located in the gas stream, either before or after the fan wheel. The function of radial built-in gas-flow straightening and re-directing vanes is to reduce energy losses due to turbulent flow and

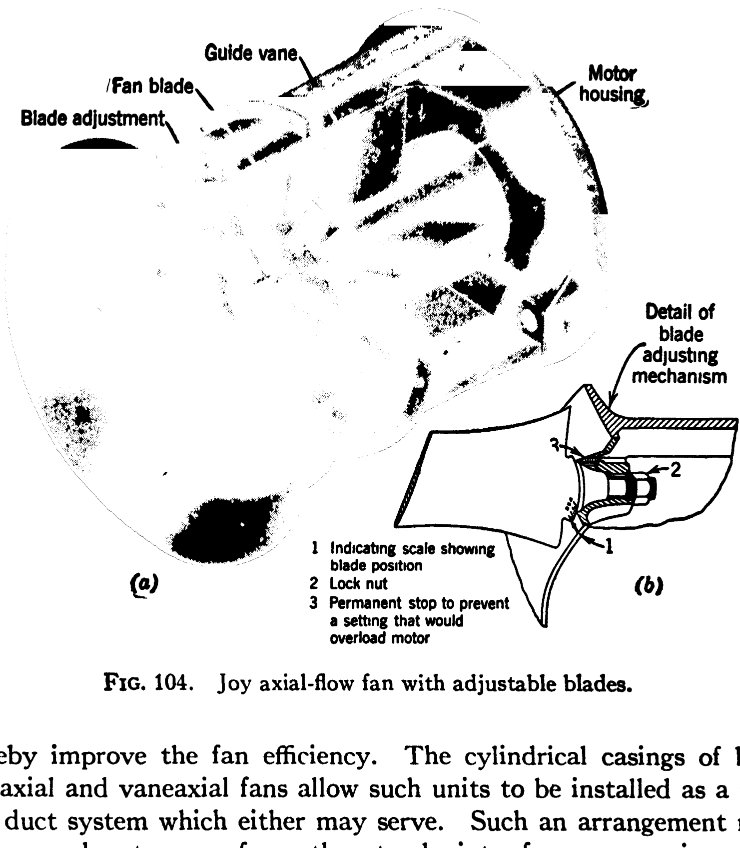


FIG. 104. Joy axial-flow fan with adjustable blades.

thereby improve the fan efficiency. The cylindrical casings of both tubeaxial and vaneaxial fans allow such units to be installed as a part of a duct system which either may serve. Such an arrangement may be very advantageous from the standpoint of space requirements. Either form of fan when attached to the inlet end of a duct system may have a flared or bell-shaped inlet which serves to reduce the entrance losses due to friction and turbulence as air flows into the fan.

The commercial axial-flow fan of Fig. 104 has adjustable blades which may be reset after installation to compensate for unexpected resistances to be overcome. Other axial-flow fans are designed so that the guide vane angles can be changed by moving an external lever. Such a

feature allows the capacity of the fan to be varied when the impeller is operated at a constant rotative speed.

164. Centrifugal Fans. The air- or gas-moving equipment of either forced- or induced-draft systems may employ centrifugal fans of the following types: (1) steel-plate, and (2) multiblade.

All centrifugal fans, Fig. 103(*d*), consist of a wheel which is rotated within a housing usually constructed of sheet metal. The housing of steel-plate and multiblade fans is of a scroll shape, as in Fig. 105, that permits the air or gas to be carried away from the periphery of the wheel with slight disturbances and power losses.

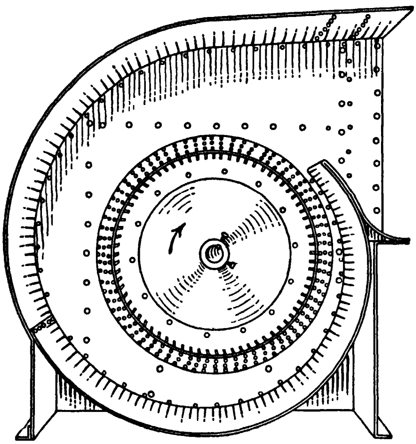


FIG. 105. Fan wheel and scroll, with side of housing removed.

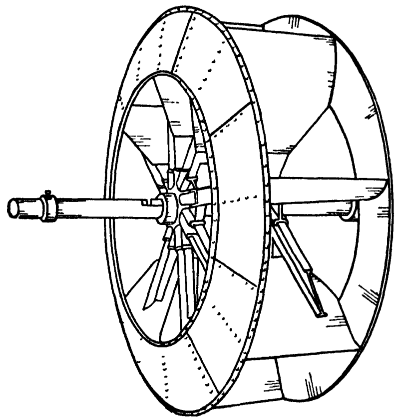


FIG. 106. Steel-plate fan wheel.

Centrifugal fan wheels have some sort of vanes or blades located at or near their rims. The action produced by a wheel in rotation is secured as the result of the tendency of the gas, adjacent to the forward sides of the fan blades, to flow radially outward. This action is due to centrifugal force, and the gas is discharged from the blade tips into the scroll of the housing. As a result of this movement a pressure less than atmospheric is produced at the center of the wheel, and a positive air pressure is built up in the scroll. Air or gas flows axially into the wheel to take the place of that discharged. Consequently, not only may centrifugal fans be used to exhaust air or gases through a duct system attached to the fan-housing inlet, but also they may be made to discharge the same air or gases under several inches of water pressure through a duct system leading away from the housing scroll.

STEEL-PLATE FANS. In fans of this type the wheels consist of one or two spiders, each having from 6 to 12 arms. Each pair of arms carries

a flat steel float or blade of some radial length as in the typical wheel of Fig. 106. The wheel floats may be straight, or may be curved either forward or backward, depending upon the operating characteristics desired. The blade curvature has a marked effect upon the performance of a given type of fan.

MULTIBLADE FANS. These fans have wheels which embody either forward-curved blades as in Fig. 107 or backward-curved blades as in Figs. 108 and 109. In all examples shown the construction involves a center plate or disk upon which the wheel hub is mounted, a shroud

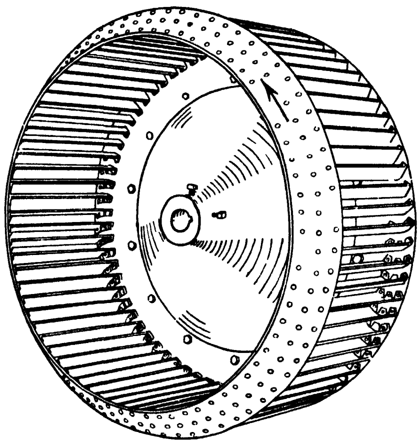


FIG 107 Forward-curved-blade fan wheel

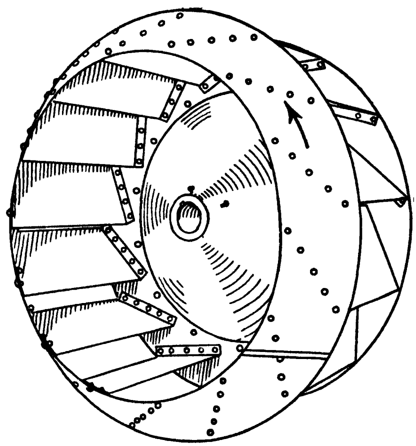


FIG 108 Backward-curved-blade fan wheel

ring at the air- or gas-inlet side of the wheel, and many blades which are uniformly spaced between the center plate and the shroud at their perimeters. Figures 107 and 108 indicate solid thin blades of curved steel plates with the wheel parts fastened together with rivets. When the wheel widths must be greater than those indicated by Figs. 107 and 108 the lateral lengths of the blades are kept short. This is accomplished by the insertion of annular rings to give strength and rigidity to the wheel assembly when it is rotated at high speeds. The method of wheel construction shown by Figs. 107 and 108 has been in use by many manufacturers for a long period of time.

The hollow backward-curved blades of the single-width single-inlet fan wheel of Fig. 109 are a development that utilizes the desirable features of an airfoil section. The hollow die-formed blades are welded to the center disk and the shroud ring to form a wheel of rigid construction without the use of either stiffening rings or rods. Either air

or gas passing through the wheel flows with small friction and turbulence losses to give high efficiency of fan operation. Another builder of fans produces hollow backward-curved blades which have a vent at the trailing edge of each blade. The wheels are in some cases constructed to have double inlets and double widths and employ a stiffening ring mid-

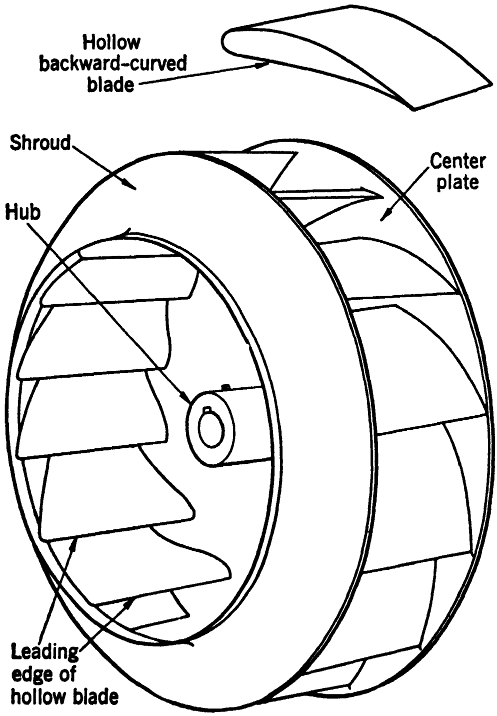


FIG. 109. Chicago Blower Corporation airfoil backward-curved-blade centrifugal fan wheel.

way between the inlets to give shorter lengths of blades and more rigid wheel construction. The housings of all multiblade fans mentioned here are of the general form shown by Fig. 105.

Another form of multiblade fan wheel employs blades that are in the form of surfaces of two tangent cones. Such construction gives a blade that has marked forward curvature at its inward edge and a definite backward curve at the exit edge. This design produces desirable operating characteristics, and a wheel may be constructed that has sufficient rigidity without the use of many stiffening rings.

Fans of the foregoing designs, when used to produce induced draft and forced draft with preheated air should be fitted with water-cooled

bearings because of the high temperatures of the gases or the air handled. Induced-draft fans of all types should be of rugged construction in order to withstand the corrosive action of the flue gases and the abrasion of ash and cinder particles discharged.

165. Effects of Fan-Blade Shapes on Air Velocities. The effects of blade shapes upon the resultant velocity of air are indicated in Fig. 110. The vector diagrams indicate velocities by the following symbols: blade-tip velocity V_t, velocity of the air flowing along the blade face V_0, and the final resultant velocity of the air V. When all the wheels shown are operated with the same blade-tip speed, the forward-curved blades give

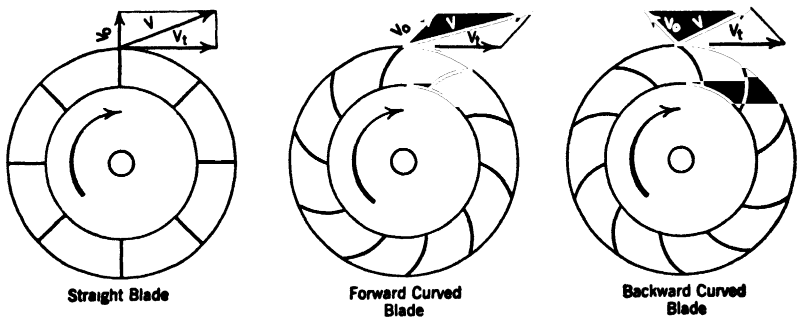
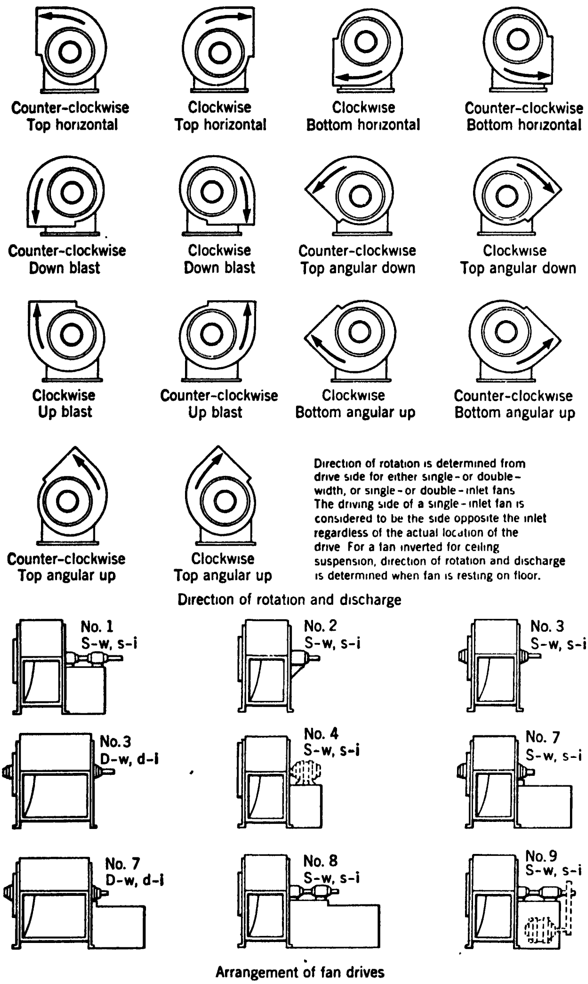


FIG. 110. Types of fan blades and resultant air velocities.

the highest and the backward-curved blades the smallest resultant air velocity. Each of the blades shown by Fig. 110 may have some modifications and retain the classification assigned to it. However, when the tip speed V_t, common to each type of blade shown by Fig. 110, is held constant, any change of either radial, forward-curved, or backward-curved blade shapes will be reflected in a change of the resultant velocity V, and in a change of the performance of the fan in which each may be used. The resultant velocity of the gas as it leaves the fan blades is of importance from the standpoint of operating speed and the noise which may be produced by the fan. Backward-curved blades in fans permit high operating speeds, high volumetric efficiencies, and a wide range of capacity at constant speed with small changes in the power requirements.

166. Fan Nomenclature. Fans of the steel-plate and multiblade types are designated according to (1) the number of inlets—single and double; (2) the width of the wheels—single and double; (3) the discharge—top, bottom, vertical, horizontal, and angular; (4) the housing —full, seven-eighths, and three-quarters; and (5) clockwise and counter-clockwise rotation as viewed from the drive side.

A full-housed fan is one in which the fan scroll is completely above the base upon which the fan rests, as illustrated by Fig. 105. Seven-eighths and three-quarters full-housed fans have the scroll extending below the top of the supporting base. More of the scroll is located below



DRIVE ARRANGEMENTS

1. S-w, s-i wheel for belt drive or direct connection. Overhung wheel. Two bearings on base.

2. S-w, s-i wheel for belt drive or direct connection. Overhung wheel. Bearings in brackets supported by fan housing.

3. S-w, s-i wheel for belt drive or direct connection. One bearing on each side and supported by fan housing. Not recommended in sizes 27-in. wheel and smaller.

3. D-w, d-i wheel for belt drive or direct connection. One bearing on each side and supported by fan housing.

4. S-w, s-i wheel for direct drive. Overhung wheel on prime mover shaft. No bearings on fan. Base mounted or equivalent for the prime mover.

7. S-w, s-i wheel for belt drive or direct connection. No. 3 plus base for prime mover. Not recommended in sizes 27-in. diameter and smaller.

7. D-w, d-i wheel for belt drive or direct connection. No. 3 plus base for prime mover.

8. S-w, s-i wheel for belt drive or direct connection. No. 1 plus base for prime mover.

9. S-w, s-i wheel for belt drive. No. 1 designed for mounting prime mover on side of base.

FIG. 111. Fan discharges and arrangements of drive.

the base line in the three-quarters full-housed than in the seven-eighths full-housed fan. The standardized designations of the rotation of a centrifugal fan and drive arrangements are shown in Fig. 111.

167. Fan Pressures. Fans in operation maintain a *total pressure* which is made up of two components, *velocity* pressure and *static* pres-

sure. Under certain conditions of operation either of these two components may be equal to zero, but both are never equal to zero when the fan is running. *Velocity pressure* is utilized in creating and maintaining the velocity of air or gas flow. *Static pressure* is the compressive pressure existing within a fluid. Static pressure is utilized in overcoming frictional and other resistance offered to the flow of air and gases. Static pressure, velocity pressure, and total pressure are all interrelated. Thus, if gas handled has a velocity increase in the duct system, part of the static pressure available at that particular locality is used to create the additional velocity pressure necessary. Likewise, if the speed of flow is reduced at some point, part of the velocity pressure at that point will be converted back into static pressure.

Any fan which develops a large static pressure in comparison with its total pressure has a greater capacity to overcome the resistances offered to air or gas flow, by duct friction, boiler-setting baffles, and fuel-bed obstructions. A fan creates pressures greater than atmospheric in a duct of the discharge side and generally produces pressures less than atmospheric in a duct on the inlet side; therefore consideration must be given to pressure measurements in both outlet and inlet ducts.

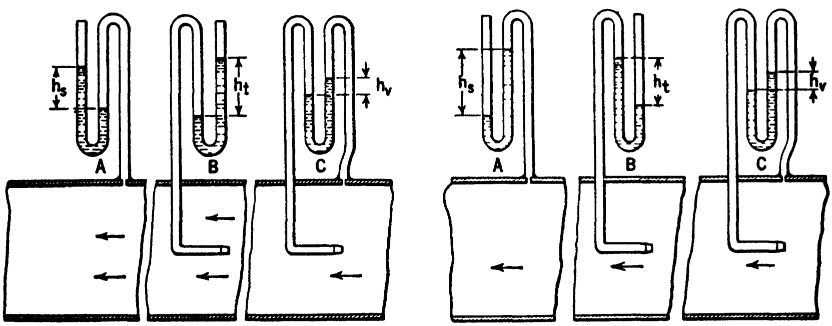


FIG. 112. Air pressure measurements.

The static pressure existing within a duct may be determined by attaching a manometer or draft gage of sufficient capacity to connections made at right angles to the longitudinal axis of the duct. These attachments should be interconnected by an external tubular ring (piezometer). Because of turbulence of air flow in ducts the piezometer-ring connection is likely to give a truer indication of static pressure than one single connection. Velocity pressure is determined by measurement of the total pressure and the subtraction of the static pressure from it, either mathematically or by the proper attachment of the static and total pressure connections to the indicating gage, as in Fig. 112.

168. Pitot Tubes. A commercial type of pitot tube as adopted by the National Association of Fan Manufacturers, Inc., and the American Society of Heating and Ventilating Engineers, Fig. 113, combines in a single instrument both static and total pressure openings. The tube illustrated is inserted in a duct with the short leg parallel to the longi-

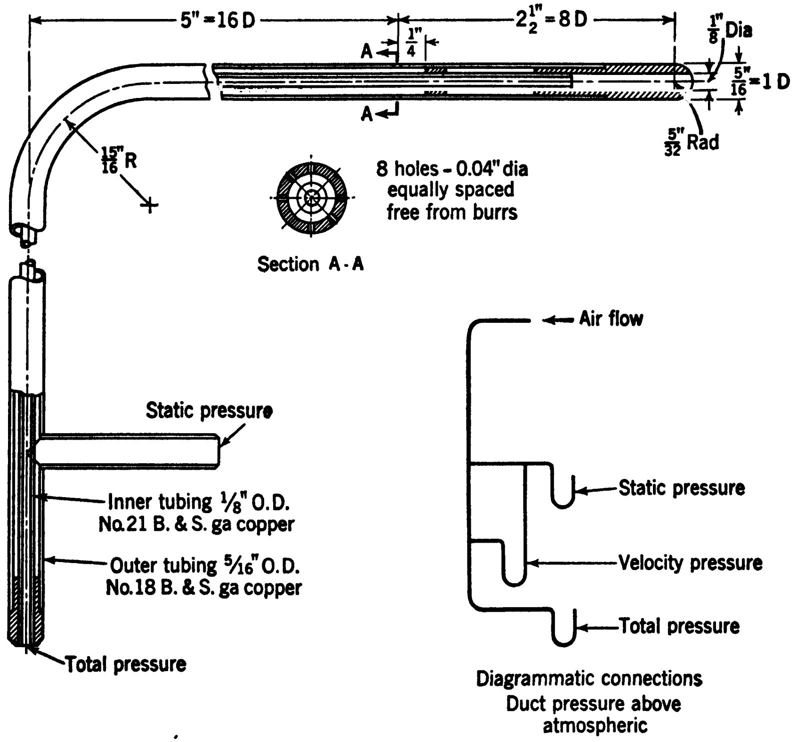


FIG. 113. Pitot tube.

tudinal axis of the duct and with the open end opposing the air stream. The open end of the short leg allows the total of the impact of the air and its static pressure to be indicated by the gage attached.

Pitot tubes are not adapted to the measurement of velocity pressures corresponding to flows less than 6 lin ft per sec unless used with very sensitive and delicate pressure-measuring gages. Pressure measurements are best made in a straight section of pipe at least 20 pipe diameters in length with at least 10 duct diameters on either side of the tube location. Erroneous and variable data are likely to be secured when observations are taken near either a fan outlet or a bend in the duct.

169. Pressures Greater and Less Than Atmospheric. Figure 112 indicates the use of a pitot tube in ducts where the air pressure is either

greater or less than that of the outside atmosphere. The static pressure is represented by h_s, the total pressure by h_t, and the velocity pressure by h_v. The velocity pressure is always positive as measured. The static and total pressures are positive when the air in a duct is greater in pressure than the outside air and negative when the reverse is true, and the readings should be so indicated on the data sheet.

Because the velocity of flow in a duct is never uniform, a number of observations are necessary in order to arrive at the average velocity of

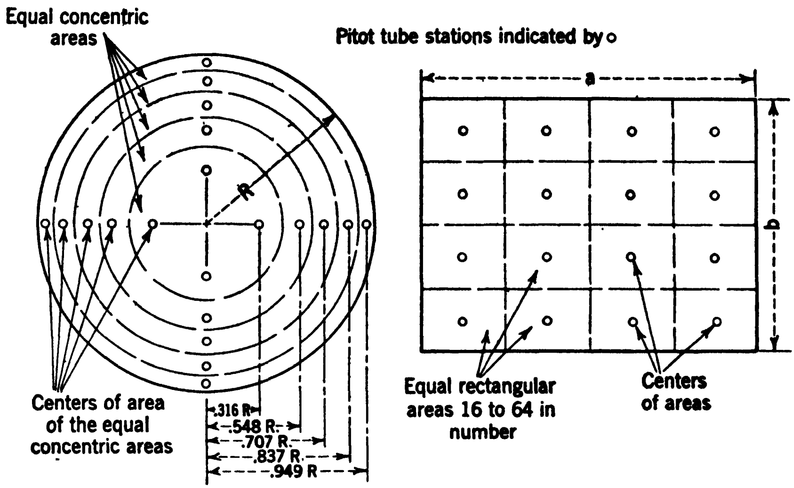


FIG. 114. Locations of a Pitot tube for a duct traverse.

flow. A scheme of locating a tube at several points on two diameters of circular ducts and in rectangular and square ducts is shown by Fig. 114. Observations taken at the points indicated serve to determine the average velocity of flow.

170. Computation of Velocity of Flow from the Velocity Pressure. The velocity of air flow is readily calculated from measured velocity pressures by the equation

$$V = \sqrt{2g \frac{Dh_v}{12d}} \times 60 \tag{192}$$

where V = velocity, fpm.

g = acceleration due to gravity, 32.17 ft per sec^2.

D = weight of 1 cu ft of water at the temperature of the gage fluid, Table XIII.

h_v = velocity pressure measured, in. of water.

d = density of the air flowing, lb per cu ft.

The average velocity of flow in a duct represents the mean of all the velocities at various places in the section of a duct. The square root of the velocity pressure is involved in the calculation of a velocity. Therefore in arriving at the average velocity of flow the average square root of the velocity pressures must be used and not the square root of the average of the velocity pressures read. Thus

$$\sqrt{h_{v_a}} = \frac{\sqrt{h_{v_1}} + \sqrt{h_{v_2}} + \sqrt{h_{v_3}} \cdots + \sqrt{h_{v_n}}}{n}$$

where n is the number of observations taken. For average velocity of flow V_a, equation 192 should be arranged as

$$V_a = \sqrt{2g \frac{D}{12d}} \times \sqrt{h_{v_a}} \times 60 \tag{193}$$

Knowledge of the relation of the average velocity in the duct to the velocity existing at the duct center is desirable. The center velocity is usually but not always greater than the average velocity of flow. The ratio is known as the *center coefficient* and is expressed as

$$C = \frac{V_a}{V_c} = \frac{\sqrt{h_{v_a}}}{\sqrt{h_{v_c}}} \tag{194}$$

where C = center coefficient.
V_a = average velocity in duct, fpm.
V_c = velocity at the center of the duct, fpm.

The center coefficient varies with the velocity of flow, the density of the gas handled, and the shape of the duct cross section. Usual values of C for ducts of circular shape range from 0.88 to 0.93, but ducts of other shapes may have coefficients greater or less than these values. When the center coefficient C has been accurately determined, only the center velocity pressure h_{v_c} need be measured, and $V_a = CV_c$.

171. Total Pressure Difference Developed by a Fan. In the operation of a fan, the gases handled arrive at the inlet to the fan housing under atmospheric pressure when no inlet duct is used and at pressures less than atmospheric when exhausting through an inlet duct. Certain types of blowers, such as gas boosters, have pressures greater than atmospheric in the inlet duct to the blower or fan.

If the fan has no suction duct the entry losses to the fan housing are considered part of the fan losses and are reflected in the mechanical efficiency of the fan. If the fan has no discharge duct leading away from its outlet the discharge static pressure is zero, and the total pressure at the discharge outlet is equal to the average velocity pressure at the fan

outlet. In any event, the average total pressure difference (total pressure) created is the average total pressure at the fan outlet minus the average total pressure at the fan inlet.

Example. A fan maintains at its outlet an average static pressure of 1.25 in. of water and an average outlet velocity pressure of 0.35 in. of water. In the suction duct near the fan the static pressure is −1.25 in. of water, and the velocity pressure is 0.25 in. of water. Find the total pressure difference developed by the fan.

Solution.

$$\begin{aligned} h_T &= (1.25 + 0.35) - (-1.25 + 0.25) \\ &= 1.60 + 1.00 = 2.60 \text{ in. of water} \end{aligned}$$

Example. A blower receives gas at 0.25 in. of water static pressure and a velocity pressure of 0.35 in. of water in the inlet duct. The discharge static pressure is 15 in. of water with a velocity pressure of 0.75 in. of water. Find the total pressure difference developed by the fan.

Solution.

$$\begin{aligned} h_T &= (15.0 + 0.75) - (0.25 + 0.35) \\ &= 15.75 - 0.60 = 15.15 \text{ in. of water} \end{aligned}$$

In the determination of total pressure it may be necessary at times to compute the velocity-pressure component by making use of a rearrangement of the equation used to compute the velocity of flow.

Example. A fan discharges 24,000 cu ft of air per min through its outlet of 12.6 sq ft area and maintains a static pressure of 5 in. of water. The air temperature is 70 F, and the barometric pressure is 29.92 in. of mercury. Find the air horsepower developed.

Solution.

$$V_a = \frac{24{,}000}{12.6 \times 60} = 31.8 \text{ fps.}$$

$$h_{v_a} = \frac{12 V_a^2 d}{2gD} = \frac{12 \times 31.8^2 \times 0.075}{2 \times 32.17 \times 62.3} = 0.226 \text{ in. of water}$$

$$h_T = h_s + h_{v_a} = 5. + 0.226 = 5.226 \text{ in. of water}$$

$$\text{ahp} = \frac{24{,}000 \times 5.226 \times 62.3}{12 \times 33{,}000} = 19.8, \text{ by equation 197}$$

172. Classes and Standard Sizes of Fans. Many types of field installations require careful selection of fans. Therefore fan standards[1] have been developed as guides in the proper selection of fans. The accepted standards are stated here as: Class I fans for 3¾ in. maxi-

[1] "Standards, Definitions, Terms, and Test Codes for Centrifugal, Axial, and Propeller Fans," by National Association of Fan Manufacturers, Inc., *Bulletin* 110, 2nd ed., copyright 1952.

mum total pressure (adopted standards), Class II fans for 6¾ in. maximum total pressure (adopted standards), Class III fans for 12¼ in. maximum total pressure (adopted standards), and Class IV fans for greater than 12¼ in. water gage total pressure (recommended practice). These classifications are shown, with special permission, in Fig. 115. Certain physical dimensions and physical data for fans are listed in the bulletin under the heading "Standard Sizes."

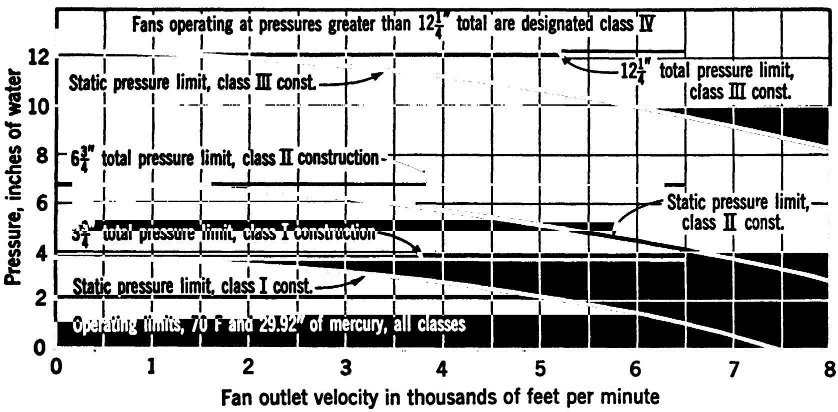


FIG. 115. Operating limits for Classes I, II, III, and IV fans as defined by the National Association of Fan Manufacturers. (Used by permission.)

173. Fan-Class Determination. The outlet dimensions of a fan of specified size fix the mean outlet velocity (fpm) for a specified volume of standard air measured in cubic feet per minute (cfm).

If the total pressure, for each class, is held constant the static pressure decreases as the fan-outlet velocity increases. Generally fans are selected on the basis of a suitable fan-outlet velocity and the necessary static pressure. When a point represented by the foregoing coordinates is located on Fig. 115 an indication of the class of fan required may be had. The curves of Fig. 115 are based on air at 70 F and a barometric pressure of 29.92 in. of mercury and having a density of 0.075 lb per cu ft. When the gas to be handled has a density d_a other than 0.075 lb per cu ft the required static pressure must be multiplied by the ratio $0.075/d_a$. The corrected static pressure together with the required outlet velocity are used to secure an indication of the class of fan required.

174. Fan Capacity and Power. The volume of gas flowing is equal to the product of the duct section in square feet and the average velocity of flow at that section.

$$Q = AV_a \tag{195}$$

where Q = cubic feet discharged per minute.
A = duct cross-section area, sq ft.
V_a = average velocity of flow, fpm.

The power output of a fan for any medium is called air horsepower (ahp) and represents work done by the fan.

$$\text{ahp} = \frac{m_a H}{33{,}000} \tag{196}$$

where m_a = weight of air handled, lb per min.
H = total head against which the fan works, ft of medium handled.

In terms of the total pressure difference created by the fan,

$$\text{ahp} = \frac{Q \times h_T \times D}{12 \times 33{,}000} \tag{197}$$

where Q = volume of air handled, cfm.
h_T = total pressure difference created by the fan, in. of water.
D = density of water, at the temperature of the gage fluid, lb per cu ft.

The power input to a fan is usually designated as shaft or brake horsepower.

175. Fan Efficiencies. The ratio of the air horsepower output to the driving power required at the fan shaft expresses the mechanical efficiency of the fan, e_m.

$$\text{Fan mechanical efficiency } e_m = \frac{\text{Air horsepower}}{\text{Horsepower input}} \tag{198}$$

Efficiencies are expressed as total and static. The total pressure of the fan is used to calculate the air horsepower when the total efficiency is desired. Static efficiency is determined by calculating air horsepower with the static pressure instead of with the total pressure.

The total mechanical efficiencies of steel-plate fans range from 40 to 60 per cent, and those of multibladed fans range from 50 to 92 per cent. Static efficiency is equal to the total efficiency times the ratio of the static to the total pressure.

176. Fan Performances. Fans should be tested according to the methods of the Standard Test Code for Centrifugal and Axial Fans as adopted by both the National Association of Fan Manufacturers, Inc., and the American Society of Heating and Ventilating Engineers. The Standard Code shows five set-ups for testing centrifugal fans and four

for axial-flow fans. In all cases pitot-tube traverses (in ducts of specific construction) are required to determine essential measurements. The required items to be secured are volumes discharged, total and static pressures, total and static efficiencies, power inputs (bhp), and speeds in rpm. Pressures are reported (in terms of in. WG) for water having a density of 62.3 lb per cu ft, and air volumes are based on standard air weighing 0.075 lb per cu ft.

For rating purposes a fan with only a discharge duct is used. Provision is made to vary the volume of the fan discharge when the fan is

TABLE X

CAPACITIES OF A BACKWARD-CURVED-BLADE FAN

No. 2¾ Clarage Class III, Type W Fan, Single Width, Single Inlet

Standard air weight of 0.075 lb per cu ft

Volume, Cfm	Outlet Velocity, Fpm	7-in. SP			7½-in. SP			8-in. SP		
		Tip Speed	Rpm	Bhp	Tip Speed	Rpm	Bhp	Tip Speed	Rpm	Bhp
10376	1600	11800	1261	16.73	12200	1303	17.92	12581	1344	19.20
11673	1800	11848	1266	18.15	12250	1309	19.45	12630	1349	20.85
12970	2000	11880	1269	19.52	12290	1313	21.00	12670	1354	22.67
14267	2200	11930	1275	21.05	12320	1316	22.19	12710	1358	24.42
15564	2400	11985	1280	22.77	12370	1322	24.60	12750	1362	26.32
16861	2600	12100	1293	24.60	12470	1332	26.38	12820	1370	28.19
18158	2800	12290	1313	26.55	12630	1349	28.16	12970	1386	30.08
19455	3000	**12580**	**1344**	**28.75**	12850	1373	30.40	13180	1408	32.20
20752	3200	12900	1378	31.04	**13150**	**1405**	**32.85**	**13430**	**1435**	**34.60**
22049	3400	13210	1411	33.25	13430	1435	35.31	13750	1469	37.40
23346	3600	13500	1442	36.20	13760	1470	38.18	14020	1498	40.20
24643	3800	13810	1476	39.12	14100	1506	41.20	14340	1532	43.35
25940	4000	14170	1514	42.27	14430	1542	44.45	14670	1567	46.65
27237	4200	14520	1551	45.75	14780	1580	48.03	15000	1603	50.10
28534	4400	14910	1593	49.50	15120	1615	51.70	15360	1641	54.00
29831	4600	15340	1639	53.45	15510	1657	55.80	15720	1680	58.10
31128	4800	15780	1686	57.90	15970	1706	60.15	16150	1725	62.50
32425	5000	16250	1736	62.45	16400	1752	64.85	16600	1774	67.60

Wheel diameter—35¾ in.; Wheel circumference—9.36 ft; Outlet area—6.485 sq ft;

$$\text{Max. bhp} = 2.35\left(\frac{\text{rpm}}{500}\right)^3$$

Data shown by bold-face type are for the most efficient operation at each static pressure listed.

operated at constant speed. The condition of most efficient operation with duct openings ranging from 0 to 100 per cent or operation with percentages of volume of wide-open discharge are determined. When the condition has been established the fan is then operated at variable speeds to secure data for the items of Table X. A fan may be operated at different speeds to create a given static pressure. In Table X data for most efficient performances are indicated by bold-faced type. From the data of the constant-speed variable outlet-opening tests, characteristic curves[2] similar to those of Figs. 116 and 117 are plotted.

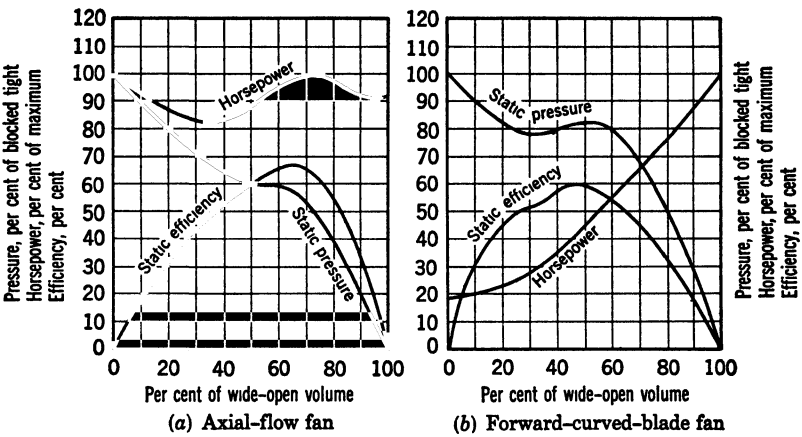


FIG. 116. Characteristic curves of fans at constant speed.

For a condition of constant speed the examples *a* and *b* of Fig. 116 are typical for the fan types involved.

Fans which produce the characteristic curves of Fig. 117, when operated at constant speed with standard air being handled, are suitable for either forced- or induced-draft service. Because the total-efficiency and the static-efficiency curves of Fig. 117 are relatively flat a line or design of fans with similar proportions, producing such results, can function with its units operating at high efficiency over a considerable range of loads. The total and static pressure curves drop sharply after 55 per cent wide-open volume is obtained. This characteristic allows little change of volume of discharge with large variation of pressure and therefore permits parallel operation of such fans, which is important when nearly constant volume is required with varying pressure. The horsepower curve rises to a peak and decreases with further increase of

[2] "Practical Considerations in Selecting Centrifugal and Axial-Flow Fans," by John W. Markert, *Heating and Ventilating*, December, 1945, pp. 80–84.

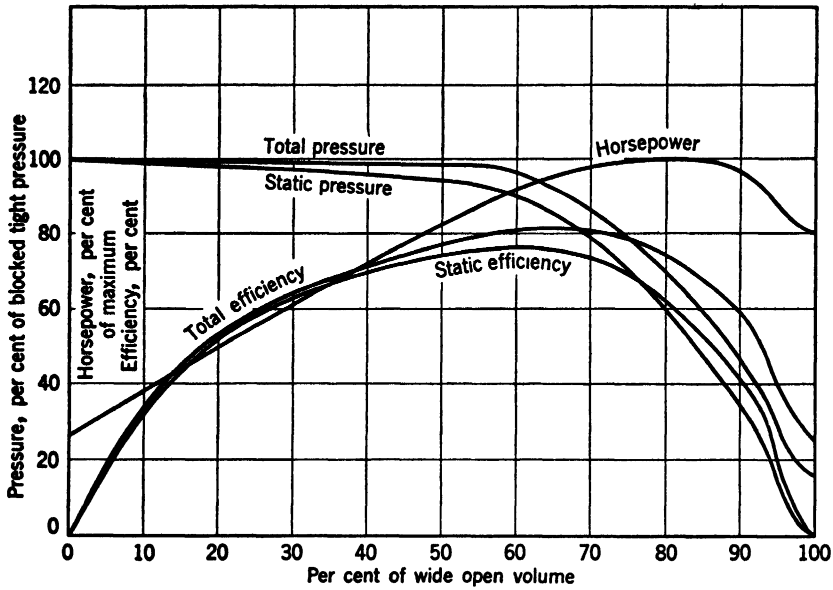


FIG. 117. Characteristic curves for a Clarage Class III, Type-W, backward-curved-blade fan.

discharge volume. Thus the fan has a nonoverloading characteristic so that, if the drive motor is selected for the maximum horsepower indicated, it is impossible to overload it if the resistance to air flow is decreased.

177. Fan Capacity and Pressure Control. The power supplied to a fan may be derived from either (1) a steam engine or a steam turbine having its speed regulation effected by hand operation of the throttle valve, (2) an adjustable-speed slip-ring motor, or (3) a constant-speed motor. Under the conditions of case 1 fan-control dampers are not necessary. When an adjustable slip-ring motor provides the power to drive a fan some sort of regulating damper at the fan outlet may be required. The electrical controls necessary to vary the revolutions per minute of constant-speed motors are expensive. Therefore, the following methods of changing the performances of fans driven by constant-speed motors are in use: (*a*) a suitable damper at the fan outlet; (*b*) either adjustable radial vanes at the fan inlet, as shown by Fig. 118, or louver inlet dampers; and (*c*) either a variable-speed hydraulic coupling or a magnetic drive. In any event the most satisfactory performances of a fan are obtained when its control is effected by regulation of the wheel speed. The least satisfactory results are derived when a constant-speed motor is used with a damper placed at the fan outlet.

Slow-speed fans may be driven through belt connections from high-speed electric motors when the ratios between the driving and the driven pulley diameters are properly selected. In such an arrangement the pressure and volume control of the fan may be secured either by an outlet damper or by radial inlet vanes or inlet louver dampers. Fans of the conoidal, radial-tipped, and backward-curved multiblade types are high-speed machines suitable for direct connection to either steam turbines or electric motors. In general, the high-speed fans are more efficient than the older slower-speed steel-plate fans.

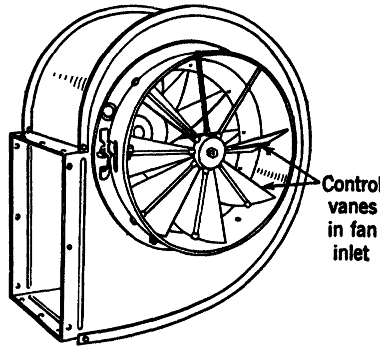


FIG. 118. Clarage fan with vortex control.

178. Fan-System Characteristics. The resistances to gas flow in a duct system must be overcome by the static pressures developed by the fan that serves it. These resistances, which vary nearly as the square of the volume of flow, are shown for one condition of operation by curve *A* of Fig. 119, which represents a system characteristic. For a given fan operated at constant rpm, speed-1 curve represents the performance of the unit as regards static pressures developed versus volumes of gases discharged. The intersection of curve *A* with speed-1 curve indicates that the duct resistance and the developed static pressure are equal and that the volume of gas handled is that indicated at *X*. If for any reason the resistances of the system are changed, its characteristic may be as shown by curve *B*. The intersection of speed-1 curve and curve *B* shows that the duct resistance and the fan static pressure are equal and that the volume of gas handled is as shown at *Y*. The latter may not be sufficient in quantity. If it is necessary to maintain the volume of flow indicated at the point *X*, when the system char-

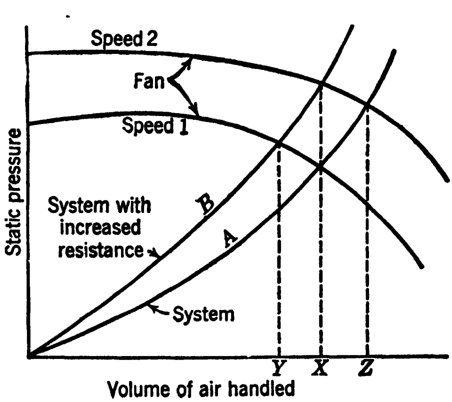


FIG. 119. Characteristics of a fan and attached duct system.

acteristic is as shown by curve B, then the fan must be speeded up to give the performances indicated by speed-2 curve. When the system characteristic is represented by curve A and the fan-speed condition of speed-2 curve exists, the volume of gas flow is that indicated at Z and the static pressure is shown by the intersection of curves A and speed-2.

179. Fan Laws. Generally basic fan laws are concerned with (1) a change of fan speed when the unit is attached to a system which has a fixed characteristic; (2) variations of the density of the gas handled; (3) an increase or decrease of the fan size, based on the maintenance of a strict proportionality of its parts, when a fixed point of rating is maintained; and (4) system resistance or static head. Other pertinent laws may be built up from considerations of the basic laws. The following statements are applicable in determining fan performances under the conditions stated.

1. Change of speed with a definite fan size, duct system, and gas density. (*a*) The capacity changes directly as the speed ratio, (*b*) the static pressure varies as the square of the speed ratio, and (*c*) the horsepower required to drive the fan varies as the cube of the speed ratio. Thus when the speed of a fan is doubled the speed ratio is 2, the capacity is doubled, the static pressure is 4 times as large and the required driving power (bhp) is 8 times that necessary before the speed was doubled.

2. Gas-density change with a fixed-size fan operated with unchanged speed to give a constant volume discharge into a fixed or constant system. Both the power required and the static pressure vary directly as the gas-density ratio.

3. Gas-density changes with a fixed size of fan operated at variable speeds to discharge a constant weight of gas into a fixed or constant system. (*a*) The volume, the static pressure, and the fan speed (rpm) vary inversely as the density ratio, and (*b*) the required driving power (bhp) varies inversely as the square of the density ratio.

4. Gas-density variation with a fixed size of fan operated at variable speed to produce a constant discharge pressure against the resistance offered by a fixed or constant duct system. The volume discharged, the fan speed (rpm), and the required power (bhp) vary inversely as the square root of the ratio of the gas densities.

5. Fan-size change with the fan operated at constant speed (rpm), constant air density, uniform fan proportions, and a fixed point of rating. (*a*) The discharge volume varies as the cube of the ratio of wheel diameters, (*b*) the static pressure varies as the square of the wheel diameters ratio, (*c*) the blade-tip speed varies as the ratio of the wheel diameters, and (*d*) the brake horsepower varies as the fifth power of the ratio of the wheel diameters.

6. Fan-size change with the fan operated with constant blade-tip speed, unchanged air density, uniform fan proportions, and a fixed point of rating. (*a*) The discharge volume and the required power (bhp) vary as the square of the wheel diameters ratio, (*b*) the discharge pressure remains constant, and (*c*) the speed (rpm) varies inversely as the ratio of the wheel diameters.

APPLICATIONS OF FAN LAWS. The following examples are indicative of some of the uses of fan laws in performance predictions.

Example. The data of Table X indicate that for maximum efficiency of operation the designated size of fan will discharge 19,455 cfm of standard air against a static pressure of 7 in. of water, when its speed is 1344 rpm, and the required brake horsepower is 28.75. Find the performance of the fan under standard conditions when the speed is changed to 1435 rpm.

Solution. The speed ratio is 1435 ÷ 1344 = 1.067. With increased speed the answers are: discharge volume, 19,455 × 1.0677 = 20,761 cfm; static pressure, $7 \times 1.0677^2 = 7 \times 1.399 = 7.98$ in. of water; and power, $28.75 \times 1.0677^3 = 28.75 \times 1.217 = 34.98$ bhp. [For the new speed of 1435 rpm and the conditions of maximum efficiency Table X listings are: volume, 20,752 cfm; static pressure, 8 in. of water; and the required power, 34.60 bhp.]

Example. The fan of the prior example is to handle air having a density of 0.0619 lb per cu ft against a static pressure of 7 in. of water when the system remains constant.

Solution. The volume, the speed, and the power vary inversely as the square root of the ratio of the densities. This means that conditions for the standard density of 0.075 lb per cu ft are multiplied by $1/\sqrt{0.0619 \div 0.075}$. In a simpler form the multiplier becomes $\sqrt{0.075 \div 0.0619}$. For actual conditions of operation the answers are: volume, $19{,}455 \times \sqrt{0.075 \div 0.0619} = 19{,}455 \times 1.1 = 21{,}400$ cfm; the necessary speed is 1344 × 1.1 = 1478 rpm; and the required power becomes 28.75 × 1.1 = 31.62 bhp.

Example. A No. 4¼ fan, with a 55¼-in. wheel diameter, has the same design, class, and relative proportions as the fan of Table X. Using performance data for the No. 2¾ fan, find the expected operating results for the larger fan when it is operated at constant tip speed, with unchanged air density, a fixed point of rating, and a static pressure of 7 in. of water.

Solution. (*a*) The ratio of the wheel diameters is 55.25 ÷ 35.75 = 1.54, and the discharge volume becomes $19{,}455 \times 1.54^2 = 46{,}150$ cfm, (*b*) the required power is $28.75 \times 1.54^2 = 68.2$ bhp, (*c*) the static pressure remains 7 in. of water, and (*d*) the speed for the larger size is 1344 ÷ 1.54 = 872 rpm. The manufacturer's listings for the larger fan, operating at maximum efficiency, are: 7 in. of water, static pressure; 46,557 cfm, volume; 871 rpm; and 68.3 bhp under standard conditions.

PROBLEMS

1. A natural-draft chimney is to produce a pressure differential of 0.95 in. of water at its base under the following conditions: outside air temperature 95 F,

average temperature of the flue gases 490 F, temperature of the draft-gage fluid 100 F, and the barometric pressure 28.85 in. of mercury. What must be the height of the chimney, in feet, if the actual draft produced is 0.85 of that theoretically possible?

2. One inch of water draft exists under the following conditions: outside-air temperature, −10 F; average temperature of flue gases, 350 F; draft-gage temperature, 65 F; and barometric pressure, 29.5 in. of mercury. What must be the height of the chimney if it produces an actual force of draft 0.8 of that theoretically possible?

3. A chimney 185 ft high may operate in either summer of winter with the following conditions: average temperature of flue gases, 500 F; temperature of draft-gage fluid, 100 F; and barometric pressure, 29.3 in. of mercury. The outside-air temperature may be −30 F in winter and 100 F in summer. How much less is the possible theoretical draft in summer than in winter? 0.75 in. WG, *Ans.*

4. A boiler plant uses 20,000 lb of coal per hr, and 14 lb of dry flue gases are produced per lb of fuel fired. The following average temperatures exist: outside air, 95 F; gases entering the chimney, 650 F; gases in the chimney, 500 F; and draft-gage fluid, 100 F. The required draft of 1.15 in. of water is 0.75 of that theoretically possible when the barometric pressure is 29.35 in. of mercury. What must be the chimney dimensions if the actual gas velocity is 0.35 of that theoretically possible? 268 ft high and 10.3 ft diameter, *Ans.*

5. A boiler plant is to use 3000 lb of coal per hr with the formation of 15 lb of dry flue gases per lb of fuel. The probable draft losses will be: through the fuel bed, 0.15; boiler passes, 0.30; straight breeching, 0.12; velocity of gas flow, 0.05; and two bends in the breeching, 0.04 in. of water each. The expected average temperatures are: outside air, 80 F; gases entering the chimney, 550 F; and average flue gases in chimney, 400 F. The actual draft may be taken as 0.8 of the theoretical, and the velocity of flow coefficient as 0.35. Find the height of and the diameter of the required chimney when the barometric pressure is 29.45 in. of mercury and the draft-gage fluid density is 62.1 lb per cu ft.

6. A fan receives air having a static pressure of 1.1 in. WG and discharges it against a static pressure of 7.5 in. WG. The inlet velocity pressure is 0.25, and the outlet velocity pressure is 0.35 in. WG. Find the total pressure created by the fan.

7. A fan operates with a static discharge pressure of 1.15 and a static inlet pressure of −0.075 in. WG. The respective velocity pressures are 0.23 and 0.15 in. WG. Find the total pressure developed by the fan.

8. Air flows in a duct with a velocity of 25 fps and a static pressure of 0.85 in. WG. The duct diameter is 4 ft, the barometric pressure 29.35 in. of mercury, and the air and the gage-fluid temperatures are 85 F. Find the total pressure of the air and the discharge in cubic feet per minute.

9. A fan fitted with only a discharge duct delivers 85,000 cfm of standard air against a total pressure of 4.24 in. WG when the static pressure is 4.0 in. WG. The temperature of the gage fluid is 70 F. The fan requires 76-bhp input. Find the total and the static efficiencies of the unit. 74.6 and 70.4 per cent, *Ans.*

10. A fan having an outlet area of 9.075 sq ft discharges 36,330 cfm of standard air against a static pressure of 12 in. WG. The gage-fluid temperature is 65 F, the fan speed 1500 rpm, and the brake horsepower input 91.50. Find the total and the static efficiencies.

11. What would be the capacity, the static pressure produced, and the power input of the fan of problem 10 if it handled standard air when operated with a speed of 750 rpm?

12. A fan is listed as having the following performances with standard air: volume discharged, 276,000 cfm; speed, 392 rpm; static pressure, 12 in. WG; and power required, 820 bhp. The system duct is unchanged, and the fan is to discharge 276,000 cfm of air at 200 F and a barometric pressure of 28.95 in. of mercury when its speed is 392 rpm. Find the static pressure required and the necessary bhp input.

9.28 in. WG and 634 bhp, *Ans.*

13. A fan operating with standard air and a system with fixed ducts gives the following performance data: discharge volume, 18,000 cfm; static pressure, 2 in. WG; speed, 583 rpm; and required power, 8.06 bhp. The fan is to be used with the system unchanged and is to handle the rated weight of air when it exists at a temperature of 180 F and a barometric pressure of 29.92 in. of mercury. Find the necessary performance data for the fan under the required conditions of operation.

14. A fan operating with a fixed or constant system handles 12,384 cfm of air weighing 0.075 lb per cu ft and discharges it against a static pressure of 10 in. WG when the fan speed is 2213 rpm and the required power is 26 bhp. If the air density is changed to 0.07 lb per cu ft and the discharge static pressure is to be held constant at 10 in. WG, what changes must be made in the operation of the fan?

15. A fan having a wheel diameter of 30 in. delivers 2070 cfm of standard air against 1 in. WG static pressure when its speed is 458 rpm and the power input is 0.52 bhp. Another fan of the same class, design, and relative proportions has a wheel 40 in. in diameter. If the blade-tip speed is the same for both fans, what performance data may be expected for the larger fan when it is operated with standard air?

1 in. WG static pressure, 3680 cfm, 0.925 bhp, and 343.5 rpm, *Ans.*

CHAPTER

10

Reciprocating Steam Engines

180. Application. The steam engine is a heat-power device that has heavy reciprocating parts which require two reversals of motion for each revolution of its crankshaft. As compared with steam turbines steam engines are slow-speed machines. Steam engines have excellent torque characteristics which allow them to start operation under heavy loads. When they are provided with suitable valve-gear mechanisms their direction of rotation can be easily reversed. The present applications are as driving units for pumps, fans, blowers, stokers, electrical generators of limited sizes, and compressors for either air or refrigerants. Units of 1000 bhp or less may give operating performances equal to or slightly better than steam turbines of the same size operating under comparable conditions.

Steam engines are unable to use the very high-pressure and high-temperature steam employed by large steam turbines. The capacity of engines in proportion to their space requirements is small. Engine thermal losses are large, and the operating back pressure cannot be reduced much below 4 in. of mercury absolute.

181. Engine Classifications. Common designations are (1) By *position of the frame and cylinder*—horizontal or vertical. (2) By *valve gear*—slide valve, Corliss, and poppet. Slide valves are further classed as D or flat, balanced, multiported, and piston. Corliss engines may have either releasing or nonreleasing valves. (3) By *steam flow in the cylinder*—counterflow and unidirectional. (4) By *number of cylinders for steam expansion*—single (simple), two cylinders (compound), three cylinders (triple expansion), and four cylinders (quadruple expansion). The foregoing numbers are the minimum. Simple engines may have more than one cylinder but the steam is not further expanded in successive cylinders. Triple and quadruple expansion engines may have more than three and four cylinders, respectively. (5) By *rotative speed*—low, medium, and high. (6) By *ratio of stroke to diameter of cylinder*—long and short. (7) By *exhaust disposal*—noncondensing and condensing. Any steam engine embodies several of the foregoing classifica-

tions, depending upon the service to be rendered and the thermal economy desired.

182. The Simple Engine. Figure 120 illustrates sectional elevations of a small vertical counterflow single-cylinder steam engine equipped with a D slide valve and a throttling governor, which is not shown.

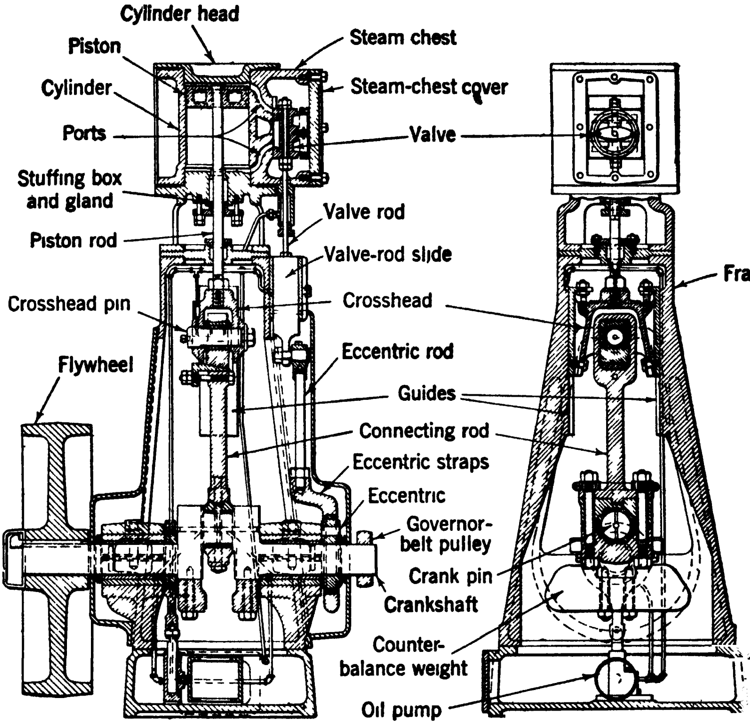


FIG. 120. Sectional elevation of a simple vertical engine.

The engine is double acting because a power cycle is produced in each end of the cylinder during one revolution of the crankshaft. The designation counterflow arises from the fact that the steam leaves the cylinder at the same end at which it entered.

Steam from the supply line is admitted by the throttle valve and passes through the governor to the steam chest. The D slide valve alternately admits steam to and releases the steam from each end of the cylinder. The valve has a reciprocating motion produced by the gear composed of the eccentric, the eccentric straps, the eccentric rod, the valve slide, and the valve rod. The purpose of steam-engine valves and the gear which actuates them is to control the distribution of steam

within the engine cylinder. As shown by Fig. 120 the valve is in a position to admit steam to the upper end of the cylinder and to allow the escape of used steam from the lower end. The piston is moved by the pressure of the steam admitted to the cylinder. At some point of the piston stroke the valve completely closes the supply port, and the flow of steam from the steam chest is cut off. After cut-off has taken place, the pressure within the cylinder decreases owing to expansion of the steam. Energy stored in the heavy flywheel during the period of steam admission is used to maintain the engine speed. Near the end of the piston stroke the valve moves to uncover the steam port of the cylinder end under consideration. When this movement takes place the expanded steam is released and flows to the exhaust passage underneath

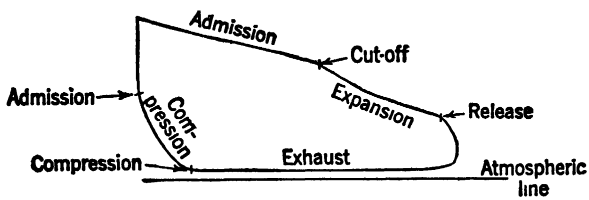


FIG. 121. Actual PV diagram.

the valve as shown for the lower (crank) end of the cylinder, Fig. 120. The exhaust of expanded steam underneath the valve occurs while live steam is being admitted to the opposite (head) end of the cylinder and continues until near the end of the return stroke of the piston. When the valve has moved to close both ports of the cylinder completely, some expanded steam is entrapped ahead of the moving piston and is compressed with a rise in pressure. This compression acts as a cushion for the returning piston and continues until the valve has moved to admit live steam to that end of the cylinder. One such complete cycle takes place in each end of the cylinder per revolution. The cycle just described can be shown graphically by means of a diagram produced by an instrument known as an indicator, when the engine is in operation. Figure 121 is such a diagram, for a throttling-governed engine, which represents the pressure-volume relations within one end of the engine cylinder at the time the figure is drawn. All ordinates of the figure represent steam pressures, and the abscissas corresponding to them may represent either the total steam volume, piston displacement, or the portion of the engine stroke completed.

The reciprocating motion imparted to the piston by the alternate admission, expansion, and release of steam in each end of the cylinder is transmitted by the piston rod to the crosshead. This part moves

back and forth between the crosshead guides and delivers its motion to one end of the connecting rod. The connecting rod is fitted at each end with a pin joint, and it converts the reciprocating motion of the crosshead to rotary motion at the crankshaft.

183. Engine Nomenclature. In steam engine parlance the following terms are used: *Head end* of the cylinder is farther from, and the *crank end* is nearer to, the crankshaft. *Dead center* is the position at which reversal of piston motion occurs at either end of the cylinder. An engine has two dead centers, that is, head end and crank end. *Stroke* is the linear distance traveled by the piston from one dead-center position to the other. The forward stroke is made between the head-end and the crank-end dead centers, and the return stroke is completed with motion in the opposite direction. Long-stroke engines have a piston travel greater than, whereas short-stroke engines have piston travels equal to or less than, the cylinder diameter. *Piston displacement* is the volume, in cubic feet, swept through by the piston during a stroke and is equal to the product of the piston area, in square feet, and the stroke length, in feet. Cylinder dimensions are expressed in inches, the bore of the cylinder being stated first and the stroke second. Thus an 8 in. by 10 in. cylinder is one having an 8-in. bore and a 10-in. stroke.

184. The D Slide Valve. The designation D is applied because of the sectional shape of the valve as shown by Figs. 120 and 123. Valves of this type, when used singly, do not permit cut-off within the engine cylinder at less than about five-eighths of the stroke without serious interference with other events of the cycle. The comparatively late cut-off thus secured limits the number of times that steam may be expanded within a single cylinder and reduces the heat economy of the engine. When the exhaust steam from a slide-valve engine may be used for feedwater heating, process work, or the heating of buildings, the over-all economy of the plant may be considerably improved.

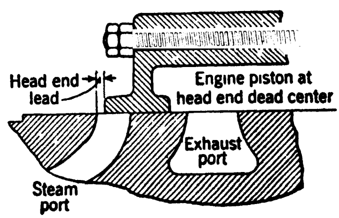


FIG. 122. Valve lead.

185. Lead and Lap. *Lead* is the amount by which a valve uncovers the steam port when an engine is at a dead-center position, Fig. 122. Lead is given to an engine to aid in cushioning the piston as it comes to rest, to reduce wire-drawing losses as the piston starts on its outward stroke, and to secure earlier release.

Steam lap is the amount by which the admission edge of the valve overlaps the edge of the port when the valve is in midposition, which is the middle of its travel, as in Fig. 123. *Exhaust lap* is the overlap of

the release edge of the valve over the release edge of the port when the valve is in the position shown by Fig. 123. The steam and exhaust laps are not usually equal for the two cylinder ends, owing to the angularity of the connecting rod and the eccentric rod. Steam laps are necessary

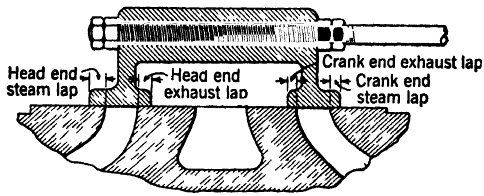


FIG. 123. Steam and exhaust laps.

in order to give cut-offs earlier than the end of the stroke; a valve without either lead or laps gives cut-off at the end of the stroke and is often a part of direct-acting pumps. Exhaust laps aid in giving compression to the steam near the end of the return stroke, although some automatic high-speed engine valves were built without exhaust lap.

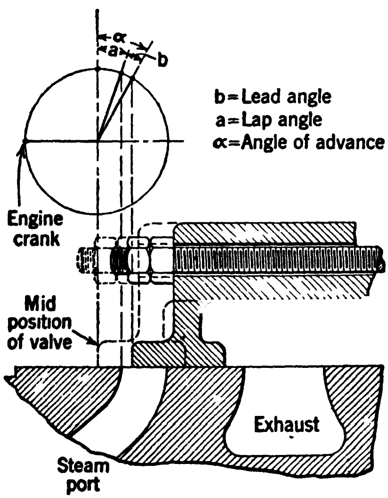


FIG. 124. Angle of advance.

186. Angle of Advance. The effect of an eccentric, Fig. 120, is that of a crank which moves the valve back and forth as the eccentric is rotated. This effect is obtained by making the eccentric in the form of an enlarged crank pin whose center does not coincide with that of the shaft upon which it is mounted. The distance between the center of the crankshaft and the center of the eccentric is called the *eccentricity* and is equal to the effective crank arm of the eccentric.

If the valve had no laps and no lead the effective crank arm of the eccentric would be set ahead of the engine crank by an angle of 90 deg. However, the valve must be moved to give the necessary lead at dead center, which necessitates that the eccentric be ahead of the crank by an angle equal to (90 deg + α), Fig. 124. The sum of the two angles a and b is the angle of advance, α, which is necessary to overcome the steam lap and to give the desired lead.

187. Balanced Valves. The steam pressure on the back of a valve may be partially relieved in several ways. The engine of Fig. 120 has a packing ring fitted to the back of the valve and held against the steam-chest cover by the action of a spring, thereby relieving much of the frictional load of the valve which is imposed on its gear. *Double-ported* slide valves have openings through them to permit the flow of steam to the engine ports. This cuts down the amount of the valve travel, and when some pressure relief on the back side of the valve is provided the frictional load of its gear may be reduced.

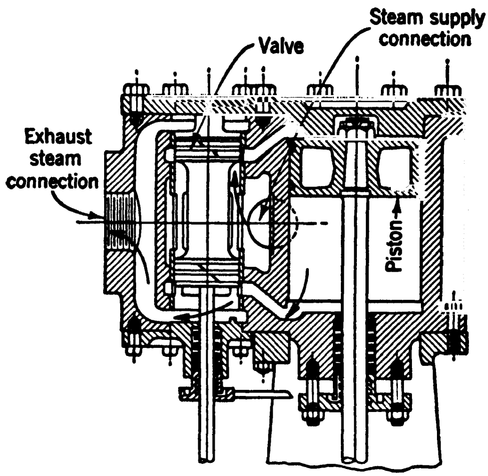


FIG. 125. Piston valve at head-end admission.

The *piston* type of valve, Fig. 125, is in principle a D slide valve. Piston valves move back and forth in a cylindrical chamber, and the steam pressure is balanced as the pressure is exerted on all of its surfaces. The flow of steam in the piston valve is indirect as compared with the flow of steam by the working edges of the D slide valve. Note in Fig. 125 that the inlet edges of the piston valve correspond to the exhaust edges of the D valve.

188. Speed Regulation. The purpose of an engine governor is either to limit or to increase automatically the power input of the unit as its load decreases or increases. The regulation must be very close in certain classes of work, such as engine-driven electrical generators where the voltage varies directly as the speed. Governors must also serve to prevent excessive and dangerous speeds. In order to give satisfactory regulation, a governor must be stable or assume a definite position for any predetermined speed. Mechanical speed regulation of steam engines is accomplished by use of either *throttling* or *cut-off* governors.

Throttling control is effected by varying the initial steam pressure within the engine cylinder. The action of a throttling governor with three different engine loads, ranging from light to heavy, is shown in Fig. 126.

Cut-off control of engine speed is accomplished by changing the volume of steam admitted to an engine cylinder as the load fluctuates. This action can be secured without a great drop of the initial steam pressure, as indicated by Fig. 127, which is for four load conditions. With full or normal loads the duration of the admission of steam to the engine cylinder is such that cut-off occurs at ¼ to ⅓ of the stroke. The point of steam cut-off comes early in the stroke of the engine with light loads and later when they increase. The various forms of cut-off governors change the point of cut-off by varying either (1) the angle of advance of an eccentric, (2) the eccentricity of an eccentric and the amount of valve travel, or (3) both the angle of advance and the valve travel.

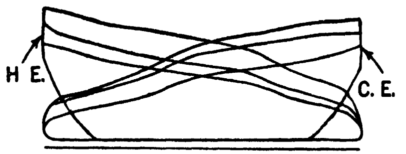


FIG. 126. Action of a throttling governor.

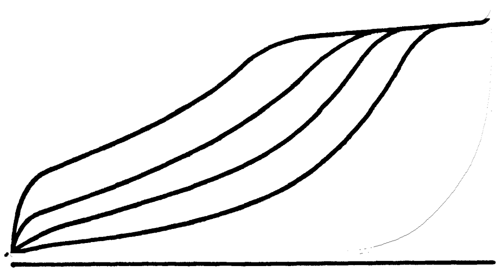


FIG. 127. Action of a cut-off governor

189. Multivalve Engines. The thermal economy of counterflow steam engines may be improved by designing them with two or more valves instead of a single valve. Such an arrangement allows independent adjustment of the valves so that earlier cut-off of the steam is possible without interference with the other events of the cycle. With earlier cut-off in the stroke the steam pressure and temperature ranges are increased as the number of steam expansions is made greater, thereby improving the thermal economy of the engine. Other advantages which accrue from the use of two or more valves are quicker action of the valve at cut-off, reduction of the length of port passages, and either the elimination or the reduction of some of the thermal losses of a steam engine

which are mentioned later. Multivalve engines discussed later are Corliss, poppet-valve, and unaflow types.

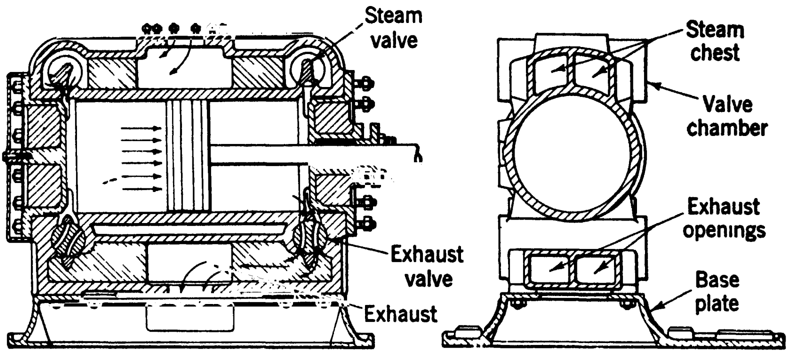


FIG. 128. Murray Corliss engine cylinder with valves.

190. The Corliss Engine. With the exception of the valves and their gear the essential features of a Corliss engine are those common to many steam engines. The valve gear is capable of cut-off governing,

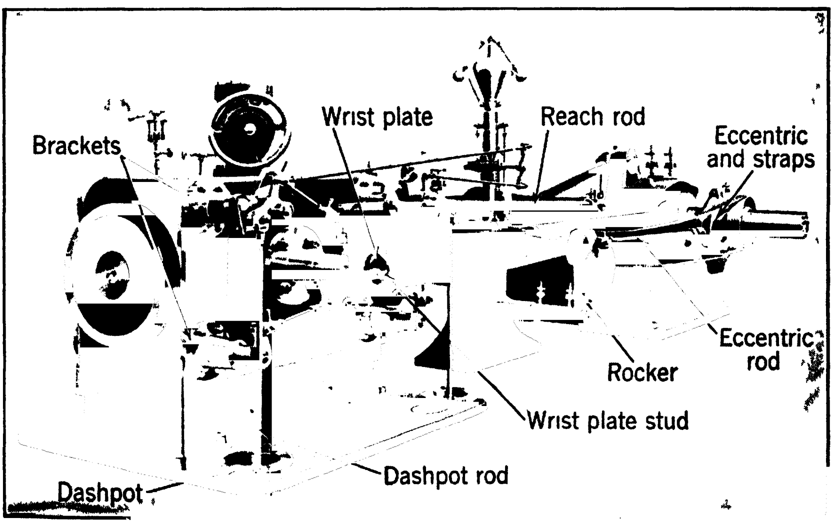


FIG. 129. Murray Corliss standard steam engine.

and the cylinder casting embodies very short port passages for both the cylindrical steam and exhaust valves. These features reduce the cylinder clearance, permit more expansions of the steam, and reduce the initial steam condensation losses below those of slide-valve engines.

A typical Corliss engine cylinder casting with its two steam valves at the top and its two exhaust valves at the bottom is shown by Fig. 128. The cylinder illustrated has its valves actuated from a wrist plate, mounted upon a stud, Fig. 129, and oscillated by an eccentric located upon the engine shaft. The motion is transmitted from the eccentric

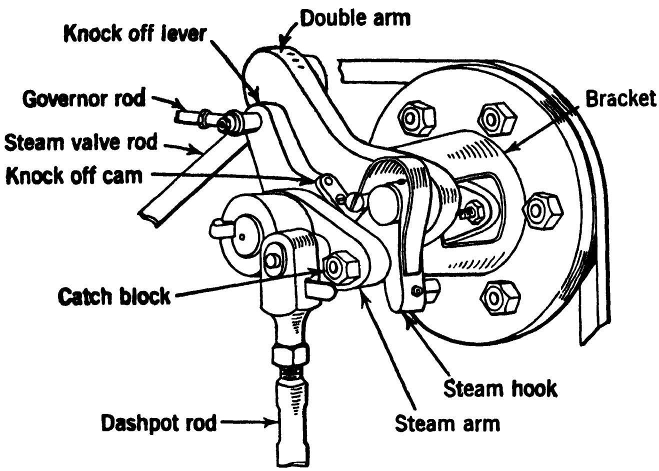


FIG. 130. Murray Corliss steam-valve gear.

by means of straps, an eccentric rod, a pivoted rocker, and a hook rod. Each valve is rotated by a T-headed spindle which has an arm keyed rigidly to it. The exhaust valves are opened and closed by the wrist plate directly. Details of the actuating mechanism for a crank-end steam valve are shown in Fig. 130. The wrist plate operates the double

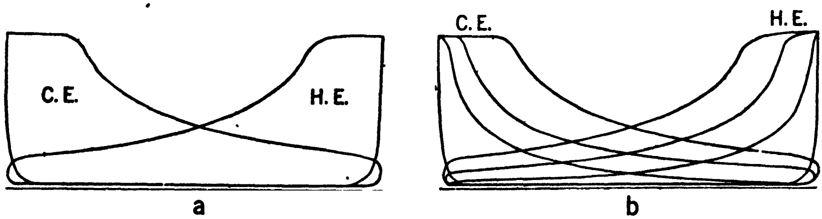


FIG. 131. Corliss engine indicator diagrams.

arms, Fig. 129, which are free to turn upon the valve spindles. Each double arm carries a steam hook which engages with the catch block of a steam arm when the double arm is brought to the proper position by the oscillating wrist plate. Rotation of the steam valve is effected by the steam arm, admission of steam to the cylinder occurs, and the dashpot

plunger is lifted against atmospheric pressure. The governor determines the position of the knockoff cam, which when it comes in contact with the steam hook disengages the steam valve from the rotating mechanism, and rapid and sharp cut-off is obtained as the dashpot plunger, under atmospheric pressure, quickly closes the steam port by rotating the valve. Figure 131 indicates the action of a Corliss engine governor with four load conditions.

191. Poppet-Valve Engines. These prime movers are adapted to the use of steam up to 400 psig and 650 F total temperature. A longitudinal section of a poppet-valve engine cylinder is shown in Fig. 132.

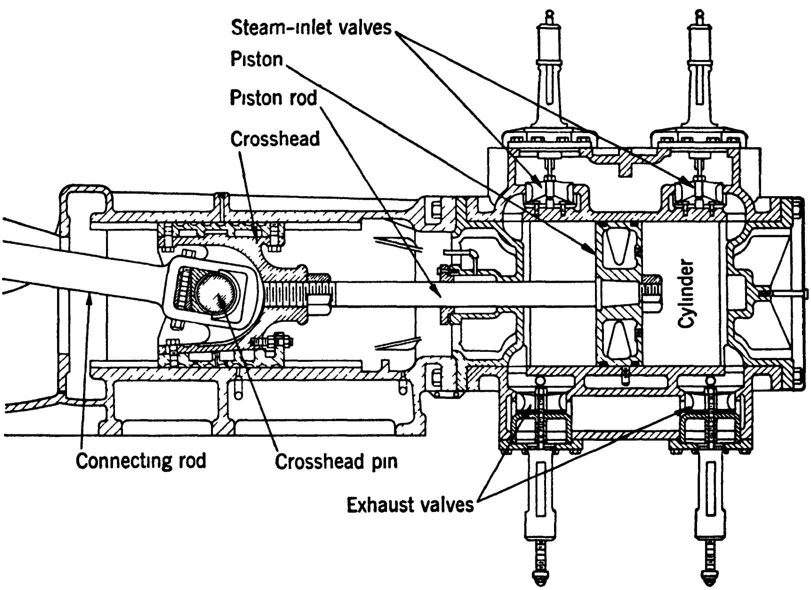


FIG 132. Erie City Lentz poppet-valve steam engine.

The steam valves at the top of the cylinder and the exhaust valves at its bottom are of the double-seat type and are operated by means of eccentrics, rods, and cams from a layshaft, along one side of the engine, that is driven from the crankshaft by bevel gears. The engine is governed by a combination centrifugal-inertia governor mounted upon and driven by the layshaft. Cut-off governing is effected by the use of slotted eccentrics to control the point of cut-off in the cylinder.

Poppet valves permit the use of highly superheated steam, as difficulties of lubrication and warping of valves are eliminated. The highly superheated steam, together with reduced clearances, minimizes the thermal losses within the engine, and good operating economy results.

192. Unaflow Engine. Large initial condensation losses may occur in counterflow steam engines, as the expanded and cooled steam chills both the cylinder walls and the steam-port passages. The single-direction or unaflow engine reduces condensation losses by use of separate inlet valves served by live-steam port passages, at each cylinder end, as shown in Fig. 133. Generally the flow of steam is in a single direction to exhaust ports located in the walls of the cylinder barrel

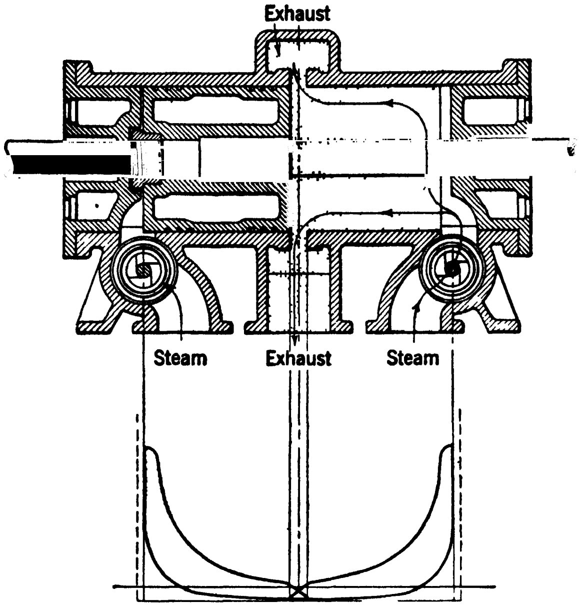


FIG 133 Diagrammatic section of a unaflow engine.

midway between its ends. The engine piston serves as an exhaust valve and alternately covers the ports for exhaust from either end of the cylinder. When operated noncondensing, unaflow engines develop a high compression pressure as the exhaust steam is entrapped during about 90 per cent of the return stroke of the piston. Noncondensing engines are often fitted with auxiliary clearances, steam ports, or automatically or mechanically operated relief valves to reduce compression pressures.

Figure 134 shows a longitudinal cylinder section of a horizontal unaflow engine with auxiliary exhaust valves for compression control. Two poppet inlet valves are provided for the admission of steam at the top of the cylinder. Each valve is driven by an eccentric mounted on a gear-driven layshaft, shown at the right of Fig. 135. The usual exhaust

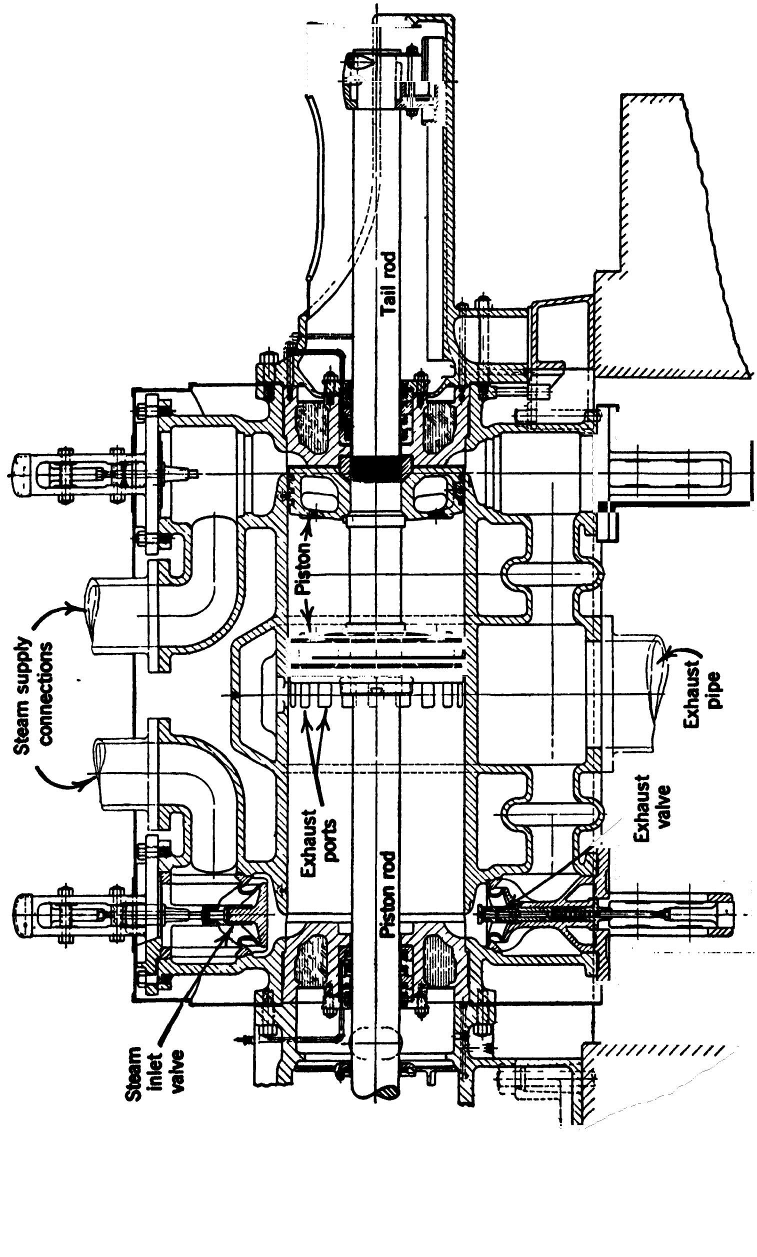


FIG. 134. Longitudinal section of a Nordberg unaflow steam-engine cylinder.

ports, opened and closed by the engine pistons, are located in the cylinder barrel midway between its ends. The piston of Fig. 133 has considerable length, whereas two pistons are used to produce the same effects in the engine cylinder of Fig. 134. As shown crank-end exhaust is occurring, and head-end admission of steam is about to take place.

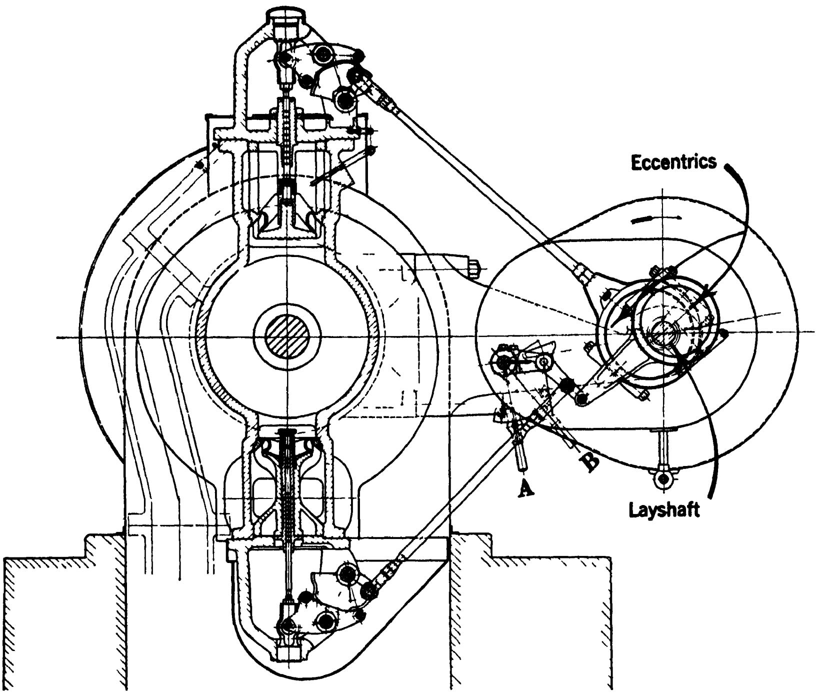


FIG. 135. Valve gear of a Nordberg horizontal unaflow steam-engine cylinder.

Compression control of the steam, during the return stroke of the piston, is effected by means of auxiliary exhaust valves shown at the bottoms of Figs. 134 and 135. A quadrant lever, which is a part of the auxiliary exhaust-valve gear, has two positions, shown by *A* and *B* in Fig. 135. When the lever is at position *A*, no lift of the poppet exhaust valves is produced by the eccentric mounted on the layshaft. Consequently there is no opening of the auxiliary valve with a reduction of the compression pressure. The setting at *B* permits the auxiliary exhaust valves to be given the maximum amount of lift and to open at the time when normal exhaust is due through the ports uncovered by the piston. With the lever setting at *B* the auxiliary valves remain open for a considerable portion of the return stroke. The compression pressure of

the steam in the cylinder can always be controlled by intermediate settings of the quadrant lever between the positions *A* and *B*. This arrangement allows the valves to open at the time for normal exhaust and to close at some intermediate point. Speed regulation of the engine is produced by means of an inertia and centrifugal governor mounted on the layshaft. The governor operates to vary the points of steam cut-off in the cylinder ends.

Unaflow engines are often more economical than small turbines operating under comparable conditions and have largely superseded the compound steam engine where variable loads imposed in engine operation occur. Attention is called to the steam consumption curves of Fig. 139; the curves for the unaflow engines show the least variation of steam consumption per unit of power developed when the load conditions are changed.

193. Compound Engines. Improvement of the thermal performance of a steam engine is possible by increasing the ranges of pressure and temperature through which the steam may be expanded within the unit. In order to effect thermal improvement, engines are built with two or more cylinders for the successive expansion of steam. The cylinder in which expansion first occurs is known as the high-pressure cylinder. Further expansion takes place in the low-pressure cylinder.

Compound engines are classified with regard to the locations of the cylinders in relation to each other and the crankshaft of the engine. The designations include (1) tandem-compound, where the pistons of the high- and the low-pressure cylinders have a common piston rod and the cylinder longitudinal axes are on the same centerline; and (2) cross-compound, cylinder longitudinal axes on parallel centerlines with the engine cranks at opposite ends of the crankshaft.

Cross-compound engines usually have overhung cranks at the ends of the crankshaft with the engine flywheel mounted between the main bearings of the engine. Ordinarily the low-pressure cylinder crank is set 90 deg ahead of the high-pressure crank, in the direction of rotation. This arrangement permits a more uniform turning effort on the crankshaft and requires a receiver. Storage space for exhaust steam from the high-pressure cylinder of a compound engine is necessary and is secured by means of receivers for the following conditions: When the exhaust of steam from the high-pressure cylinder cannot pass continuously to the low-pressure cylinder and to the engines having the low-pressure crank ahead of the high-pressure crank and requiring admission of steam to the low-pressure cylinder, when the high-pressure cylinder is not exhausting.

Engines expanding steam in four successive cylinders or stages

(quadruple expansion) are not used at the present time. Triple-expansion engines expand the steam in three or more cylinders. It is not uncommon to find triple-expansion engines fitted with five cylinders: one for the partial expansion of high-pressure steam, two intermediate cylinders, and two low-pressure cylinders.

194. Indicated Horsepower. The power (Art. 12) produced within the ends of a double-acting steam cylinder is expressed as

$$\text{ihp, head end} = \frac{p_m l A N}{33{,}000}$$

$$\text{ihp, crank end} = \frac{p'_m l A' N}{33{,}000}$$

where p_m = head-end mep, psi.
p'_m = crank-end mep, psi.
l = length of engine stroke, ft.
A = net area of piston, head end, sq in.
A' = net area of piston, crank end, sq in.
N = revolutions per minute.

When a piston rod is attached to the piston in either end of the cylinder, its cross-sectional area must be deducted from the gross piston area to give the net effective piston area.

$$\text{Total ihp} = \frac{p_m l A N}{33{,}000} + \frac{p'_m l A' N}{33{,}000} \tag{199}$$

When engine cylinders are single acting, power is not developed in the crank end, and the indicated horsepower is that of the head ends alone. The total indicated horsepower of engines having more than one cylinder is the sum of the indicated horsepowers of all the cylinders.

195. Brake Horsepower. The actual power output delivered by the crankshaft or at the rim of the flywheel of an engine is the brake horsepower. On account of friction and windage losses occasioned by the moving parts of the engine, the brake horsepower delivered is always less than the indicated horsepower developed.

From Art. 12,

$$\text{bhp} = \frac{2\pi r w N}{33{,}000}$$

where r = length of brake arm, ft.
w = net load at the end of the brake arm, lb.
N = speed of the engine, revolutions per minute.

Example. A steam engine equipped with a Prony brake having an arm $5\frac{1}{4}$ ft long ran at 200 rpm. The net load at the end of the arm was 125 lb. Find the brake horsepower.

Solution.
$$\text{bhp} = \frac{2 \times 3.1416 \times 5\frac{1}{4} \times 125 \times 200}{33{,}000} = 25$$

196. Friction Horsepower. The loss of mechanical power within an engine is known as friction horsepower. This quantity is equal to the difference of the total ihp and the bhp. Friction horsepower varies with the percentage of rating developed, size, type, rotative speed, and mechanical construction of an engine. In general, the percentage of friction horsepower is greater in multicylinder engines than in simple engines.

197. Mechanical Efficiency. The ratio of the engine power output to the power developed within the cylinder is the mechanical efficiency of an engine.

$$\text{Mechanical efficiency, } e_m = \frac{\text{bhp}}{\text{ihp}} \qquad (200)$$

The mechanical efficiencies of steam engines at normal loads range from 85 to 95 per cent, depending upon the design and workmanship of the engine. Poorly constructed engines may have 85 per cent efficiency, and high-grade machines may give 90 per cent. The mechanical efficiency of an engine varies with the load upon the engine, increasing with the load, from light loads to a maximum, and then decreasing with a further increase of engine load. Variations of friction horsepower and mechanical efficiency for a steam engine operating at different fractions of rated power are shown in Fig. 136.

198. Theoretical Thermal Economy of a Steam Engine. The efficiency of the Carnot cycle cannot be exceeded by any heat engines working between given temperature limits. As a measure of thermal performance the Carnot cycle cannot be applied to a steam cycle as the conditions of its operation cannot even be approximated. The steam engine is a machine which receives the working medium under definite conditions and which exhausts it at a definite pressure; theoretically isentropic expansion of the steam is desirable. The Rankine cycle is generally used to express the ideal performance of either a steam engine or a steam turbine operating in conjunction with other pieces of power plant equipment, such as the condenser, the feed pump, and the boiler, Fig. 13. The efficiency of operation thus determined is not the over-all plant efficiency computed by dividing the heat equivalent of the plant output by the heat energy carried by the fuel burned in the boiler furnaces. When an engine or a turbine alone is considered the initial condi-

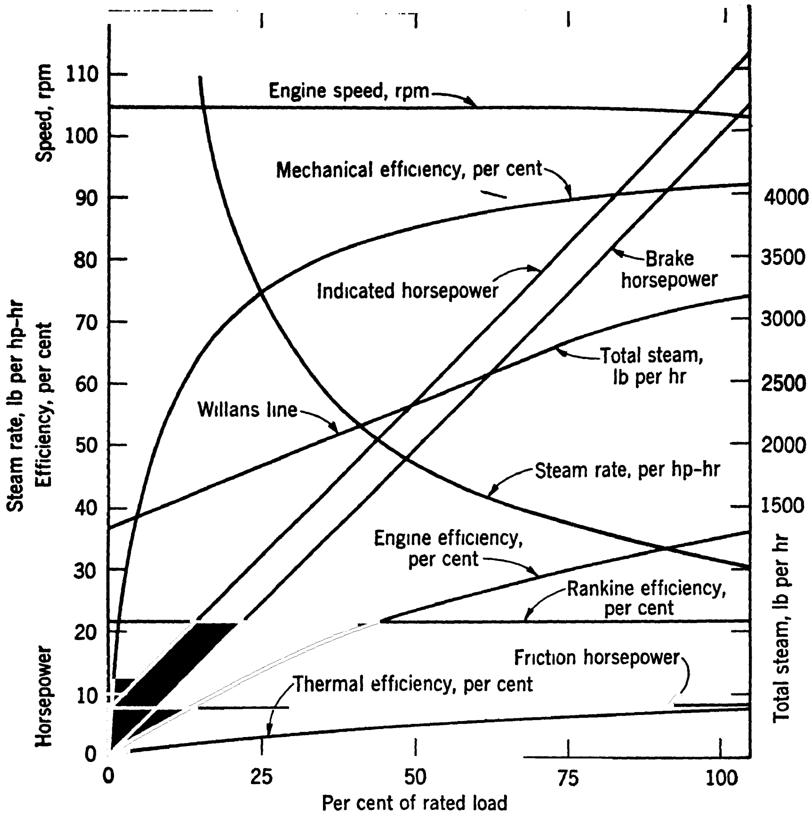


FIG. 136. Typical performance curves of a steam engine.

tions of the working substance are not duplicated after it has passed through the primé mover.

For complete expansion of steam in the Rankine cycle the pressure-volume diagram, Fig. 137, and the temperature-entropy diagram, Fig. 138, apply. Consider 1 lb of dry saturated steam entering the prime mover at a with a pressure of p_1 psia and a total enthalpy of h_a Btu per lb. Isentropic expansion (constant entropy) occurs between a and b where the pressure is p_2 psia. Condensation of the expanded steam takes place at the constant pressure p_2, and the volume of the now saturated liquid becomes $v_{f,c}$ at c. The feed pump then exerts a pressure, somewhat greater than the boiler pressure, on the liquid to discharge it into the boiler at the pressure p_1 shown at d. The pump action is an isentropic one, and the fluid enters the boiler in a nonsaturated condition with an enthalpy of h_d. The enthalpy is increased in the boiler at

constant pressure p_1, and the fluid again becomes saturated at e. Further change of enthalpy continues until the state at a is reached and the cycle $abcdea$ is completed.

On the Ts diagram, Fig. 138, the change of enthalpy in the boiler is represented by $deafg$ which is equivalent to $Q_1 = h_a - h_d$. The heat absorbed by the condenser is shown as $bfgc$ and is $Q_2 = h_b - h_c$. The net work of the cycle is $abcdea$ and in terms of heat units is $Q_1 - Q_2 = (h_a - h_d) - (h_b - h_c)$ or $(h_a - h_b) - (h_d - h_c)$. The work done by the

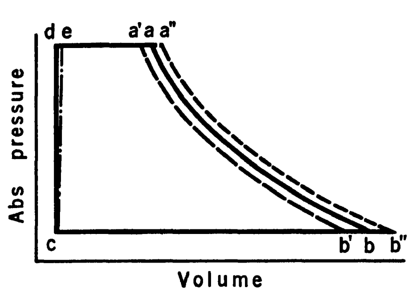


FIG. 137. Rankine cycle with complete expansion.

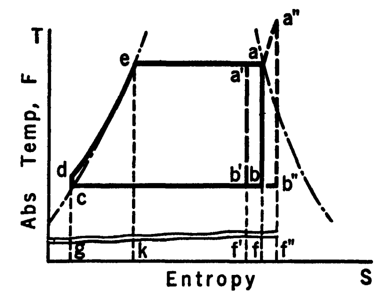


FIG. 138. Temperature-entropy diagram for the Rankine cycle with complete expansion.

pump, in Btu per pound of steam, is $W = (h_d - h_c)$. Then $h_d = h_c + W$, and $Q_1 = h_a - h_d = h_a - h_c - W$. The efficiency of the ideal Rankine cycle is $(Q_1 - Q_2) \div Q_1$, or

$$e_R = \frac{(h_a - h_b) - (h_d - h_c)}{(h_a - h_c) - (h_d - h_c)} \tag{201}$$

and

$$e_R = \frac{(h_a - h_b) - W}{(h_a - h_c) - W} \tag{202}$$

If the fluid at c, having a volume of $v_{f,c}$ cu ft per lb, were incompressible and the discharge pressure of the pump were p_1, then $(h_d - h_c)$ would be equal to $144\ (p_1 - p_2)v_{f,c} \div 778.26$. The enthalpy of the liquid of compressed water is discussed in Art. 66. When the boiler pressures are low, say below 400 psia, $(h_d - h_c)$ may be ignored, but in high-pressure plants it may be desirable to give consideration to the work done by the feed pump. For wet saturated steam the points c', d', and f' apply, and for superheated steam the points c'', d'', and f'' are applicable; these symbols are to be inserted in the foregoing discussion instead of c, d, and f.

For the Rankine engine cycle the usual expression of efficiency is

$$e_R = \frac{h_a - h_b}{h_a - h_c}$$

where the symbols have the same significance as in equation 202. However, in a steam power plant the flow of steam from the outlet of the steam generator to the throttle of the steam engine is always accompanied by heat losses and a drop in pressure. Consequently, the steam entering the prime mover has an enthalpy h_1, which is less than the enthalpy h_a of the steam leaving the steam generator. When this fact is recognized the expression for the efficiency of the Rankine cycle becomes

$$e_R = \frac{h_1 - h_2}{h_1 - h_{f,2}} \tag{203}$$

where h_1 = enthalpy of the steam (dry saturated, wet saturated, or superheated) at the throttle pressure of p_1, Btu per lb.

h_2 = enthalpy of the steam after isentropic expansion from the initial pressure of p_1 to the exhaust pressure of p_2, Btu per lb.

$h_{f,2}$ = enthalpy of the liquid at the exhaust pressure p_2, Btu per lb.

The common practice is to consider that the enthalpy of the liquid at the exhaust pressure p_2 would be used again if the exhaust steam were converted into feedwater at the temperature corresponding to the absolute exhaust pressure. Therefore the heat supplied to the engine is $h_1 - h_{f,2}$.

The ideal steam consumption of an engine, pounds per horsepower-hour, is

$$m_s = \frac{2545}{h_1 - h_2} \tag{204}$$

where h_1 and h_2 are as defined in equation 203. If the ideal steam consumption per kilowatt-hour is desired the numerator of equation 204 becomes 3413.

Example. Find the Rankine cycle efficiency and the ideal steam consumption per horsepower-hour for a steam engine operating with saturated steam initially dry at 160 psia and exhausting against a back pressure of 2 psia.

Solution. From Table XV the total enthalpy of the initial steam is $h_1 = 1195.1$ Btu per lb. The quality of the steam after isentropic expansion from 160 to 2 psia is $x_2 = (1.564 - 0.1749)/1.7451 = 0.796$. The total enthalpy after expansion with constant entropy to an absolute pressure of 2 psia is $h_2 = 93.99 + (0.796 \times 1022.2) = 907.7$ Btu per lb. $e_R = (1195.1 - 907.7)/(1195.1 - 93.99) = 0.261$ or 26.1 per cent. The theoretical steam consumption per horsepower-hour is $2545/(1195.1 - 907.7) = 8.8$ lb.

199. Total Steam Consumption. Dependent upon the load carried, a steam engine requires a certain total amount of steam per hour which includes not only that used inside the cylinders but also that used in cylinder jackets and reheater coils. The total weight of steam necessary per hour may be shown graphically by plotting the total amounts used per hour as ordinates against the corresponding engine loads (either as ihp, bhp, or kw) as abscissas. The shape of the resulting curve, which is known as the *Willans line*, is dependent upon the method by which the engine is governed. For throttling-governed engines the curve is nearly a straight line until the engine is overloaded. Engines controlled by automatic cut-off governors usually give curves which are bowed downward with respect to the axis of the abscissa. Such curves when partially completed are of some value in predicting the total steam consumption for loads not already measured.

200. Specific Steam Consumption or Steam Rate. The weight of steam supplied to a steam engine per horsepower per hour is the *steam rate* of the engine. The expression is made on the basis of either brake or indicated horsepower. The steam rate of an engine is dependent upon a number of factors: (1) initial steam pressure, (2) initial steam temperature, (3) exhaust pressure, (4) number of expansions of the steam, (5) number of cylinders used for expansion, (6) method of governing, (7) valves and valve gear, (8) percentage of rating developed, (9) type of engine—counterflow or unaflow, (10) mechanical design and construction, (11) cylinder insulation, and (12) steam jackets and reheaters.

Comparative performances of steam engines cannot be judged by the steam rates alone because of the number of factors affecting the engine operation. The relative merits of thermal performance of various engines may be obtained more accurately by comparisons of the amounts of heat supplied to the engines per horsepower-hour. Because of the great diversity of operating conditions the dry steam consumptions of only a few representative engines are given in Fig. 139 for various initial and exhaust steam pressures.

201. Thermal Efficiency. Thermal efficiency is the ratio of the output (heat equivalent) to the quantity of heat supplied to the engine to produce the power. The ratio or thermal efficiency for saturated steam when computed on the basis of one horsepower is

$$e_t = \frac{2545}{m_s(h_{f,1} + x_1 h_{fg,1} - h_{f,2})} \tag{205}$$

where e_t = actual thermal efficiency.

2545 = heat equivalent of one horsepower-hour, Btu.

m_s = actual pounds of steam supplied per horsepower-hour.

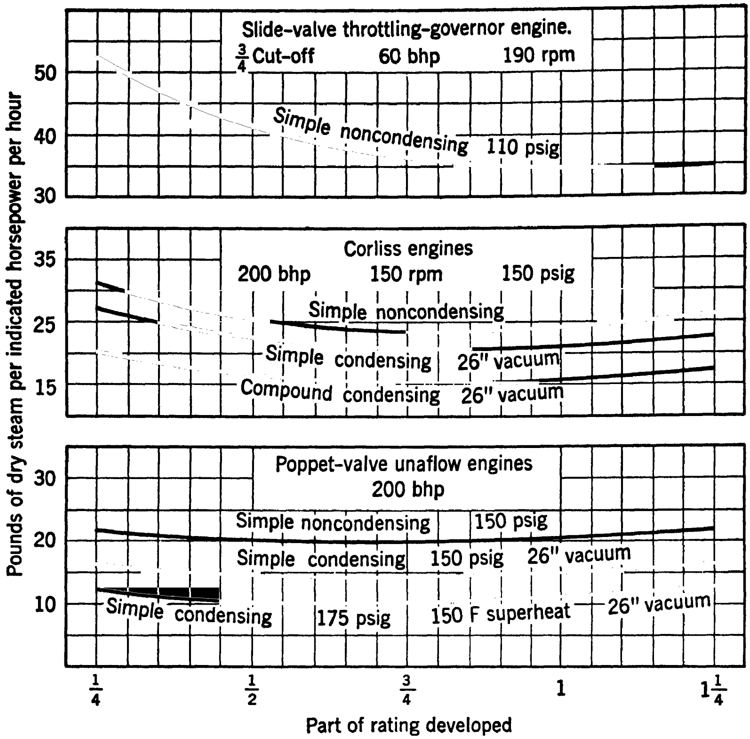


FIG. 139. Steam consumption of various engines.

$h_{f,1}$ = enthalpy of liquid at initial pressure absolute, Btu per lb.
x_1 = quality of the steam supplied.
$h_{fg,1}$ = enthalpy of evaporation at initial pressure absolute, Btu per lb.
$h_{f,2}$ = enthalpy of liquid at exhaust pressure absolute, Btu per lb.

When superheated steam is supplied to the engine the expression becomes

$$e_t = \frac{2545}{m_s(h - h_{f,2})} \tag{206}$$

where h = total enthalpy of superheated steam at initial pressure absolute, Btu per lb.

The enthalpy of the steam supplied to the engine above the enthalpy of the liquid corresponding to the temperature at exhaust pressure is considered chargeable to the engine.

Inspection of equations 205 and 206 reveals that, in either case, when the initial conditions are unchanged, anything which reduces the quan-

tity m_s increases the thermal efficiency. The thermal efficiency varies inversely as the steam consumption per horsepower-hour.

Example. A steam engine uses 40 lb of wet steam per brake horsepower-hour. Initial pressure of the steam is 150 psia, exhaust pressure 15 psia, and steam quality 0.98. Find the thermal efficiency.

Solution.

$$e_t = \frac{2545}{40[330.51 + (0.98 \times 863.6) - 181.11]} = 0.064 \quad \text{or} \quad 6.4 \text{ per cent}$$

202. Engine Efficiency. The ideal or Rankine cycle has been set up as the best possible performance for given conditions. The theoretical efficiency for this cycle was discussed in Art. 198. The actual engine performance falls short of the ideal because the conditions of the ideal cycle cannot be fulfilled. The ratio of the actual thermal efficiency to that of the Rankine cycle for the same pressure and temperature conditions is the *engine efficiency*, e_e.

$$e_e = \frac{e_t}{e_R} = \frac{2545}{m_s(h_1 - h_{f,2})} \times \frac{h_1 - h_{f,2}}{h_1 - h_2}$$

$$e_e = \frac{2545}{m_s(h_1 - h_2)} \tag{207}$$

Where h_2 is the total enthalpy per pound of steam after it has been expanded isentropically from the initial to the final pressure. This enthalpy may be either obtained from the Mollier diagram or calculated as $h_2 = h_{f,2} + x_2 h_{fg,2}$. The final quality of the steam is computed by the method of Art. 64. The engine efficiency varies from 0.35 for poorly designed and operated engines to 0.70 or 0.80 for high-class, high-efficiency engines.

203. Thermal Losses. Certain losses occurring in steam engines affect the steam consumption and the thermal efficiency under operating conditions. These losses include (1) enthalpy of exhaust steam, (2) initial condensation, (3) re-evaporation, (4) incomplete expansion, (5) leakage, (6) radiation, (7) clearance volume, and (8) wiredrawing.

204. Improvement of Steam Economy. During a long period of time steam engines were the principal source of power in an era of great industrial development. Therefore, many attempts were made to reduce the thermal losses listed in Art. 203. In general anything which either increases the pressure and temperature range of the steam-engine cycle or produces reduction and elimination of thermal and friction losses is desirable in the improvement of engine economy and efficiency.

Improved designs and the utilization of auxiliary equipment have

somewhat lessened the amount of heat lost in exhaust steam, but at best the wasteful loss may be only 70 per cent instead of more than 90 per cent of the total heat of the steam supplied to the engine. Initial condensation (and its subsequent partial re-evaporation) by chilled port and cylinder walls involves a large thermal loss within an engine. Aids in the reduction of such losses are increased steam pressures, superheated steam, four-valve engines of the Corliss type, poppet-valve engines with superheated steam, and unaflow cylinders.

Suitable valve gears have permitted a greater number of steam expansions within a cylinder without interference of the cycle events. *Engine clearance* (in either cylinder end) is the port volume from the valve face to the cylinder plus the volume between the piston at dead-center position and the cylinder head. Some clearance is necessary in a steam-engine cylinder, but large clearance spaces reduce the engine economy because of the work necessary for the compression of steam at the end of the return stroke of the piston. A reasonable amount of clearance-space steam compression is an aid in cushioning the reversal of direction of motion of the reciprocating parts at each dead-center position. Reduction of the clearance space lessens the weight of steam necessary to fill the cylinder at the point of cut-off. When the compression of the steam within a clearance space is excessive, the temperature of the steam may rise above that of the surrounding wall surfaces and a thermal loss similar to initial condensation may occur.

Good features of engine design have lessened the amount of steam wiredrawing in port passages. Wiredrawing is a throttling action whereby the pressure of steam is reduced, its entropy is increased, and its available energy is decreased. The same sort of performance occurs in the throttling type of steam-engine governor as steam flows by the partially opened valve in its body.

PROBLEMS

1. Find the friction horsepower and the mechanical efficiency of a 12 in. by 15 in. steam engine which gives the following test data: area of the head-end diagram, 1.45 sq in.; area of the crank-end diagram, 1.25 sq in.; average length of diagrams, 3 in.; engine speed, 240 rpm; indicator-spring scale, 80 psi; diameter of the piston rod, 2½ in.; brake tare weight, 80 lb; brake gross load, 620 lb; and length of brake arm, 2.63 ft.

2. A 15 in. by 21 in. steam engine with a piston rod 2¾-in. in diameter operates at 120 rpm to give the following data: area of the head-end diagram, 3.45 sq in.; area of the crank-end diagram, 3.75 sq in.; average length of diagrams, 3.5 in.; spring scale, 60 psi; length of brake arm, 63 in.; and net load at the end of the brake arm, 1006 lb. Find the friction horsepower and the mechanical efficiency of the engine.

15.8 fhp and 88.5 per cent, *Ans.*

3. A 16 in. by 14 in. double-acting simple steam engine with a piston rod 3 in. in diameter operates at 250 rpm to give an output of 200 bhp when 225 ihp are developed within the cylinder. (*a*) Find the friction horsepower and the mechanical efficiency of the engine. (*b*) The engine-indicator diagrams average 4 in. long and are taken with a spring having a scale of 80 psi. What are the diagram areas if 5 ihp more are produced within the head end than in the crank end of the engine cylinder?

4. An 8 in. by 8 in. double-acting simple steam engine with a piston rod $1\frac{1}{2}$ in. in diameter is rated at 35 bhp when operated at 350 rpm with an initial steam pressure of 250 psia. The engine valve setting is such that equal amounts of power are developed within the two ends of the cylinder to give a mechanical efficiency of 87.5 per cent when the rated power is produced. (*a*) Find the actual mean effective pressure developed within each cylinder end. (*b*) Find the area of each indicator diagram if they average 2 in. long when taken with a spring having a scale of 160 psi. (*c*) Find the net load necessary at the end of a Prony brake 63 in. long.

5. A steam engine develops 100 bhp under the following conditions: initial steam pressure, 135.5 psig; barometric pressure, 29.6 in. of mercury; initial quality of steam, 0.99; exhaust pressure, atmospheric; and total wet steam used per hour, 3450 lb. Find the engine steam rate and the thermal efficiency.

6. A steam engine operates with steam at 140 psia initial pressure and a total temperature of 400 F, a back pressure of 5 psia, a load of 350 ihp, and a total steam consumption of 6950 lb per hr. Find the steam rate and the thermal efficiency based on indicated horsepower. 19.85 lb per hr and 11.75 per cent, *Ans.*

7. A steam engine operates with 500-bhp output, a steam consumption of 15 lb per bhp-hr, an initial steam temperature of 450 F, an initial steam pressure of 200 psia, and an exhaust pressure of 3 in. of mercury abs. Find the thermal efficiency based on brake horsepower, the Rankine efficiency, and the engine efficiency.

8. A steam engine delivers an output of 300 hp under the following conditions: initial steam pressure, 180 psia; exhaust pressure, 2.46 psia; initial steam temperature, 400 F; mechanical efficiency, 88 per cent; and steam consumption, 12 lb per ihp-hr. Calculate the thermal efficiency based on brake horsepower, the Rankine efficiency, and the engine efficiency. 16.8, 26.1, and 64.4 per cent, *Ans.*

CHAPTER

11

Steam Turbines

205. Types and Principles. Turbines are steady-flow machines in which steam enters nozzles and expands to a lower pressure. In so doing the steam jet develops a high velocity. Part of the kinetic energy of this jet can be delivered to the blades of the turbine in the same manner that a jet of water can deliver energy to the buckets of a water wheel. Those turbines which utilize the impulse of the jet on moving blades are called *impulse turbines*, Fig. 140. In this arrangement the nozzle is stationary and mounted in the frame. It is possible to shape the spaces between the rotating blades so that they constitute nozzles. The reaction on the nozzles from the issuing steam rotates the wheel. This principle identifies a pure *reaction turbine*. These two types are direct applications of Newton's law of motion, which states that for each action there is an equal and opposite reaction.

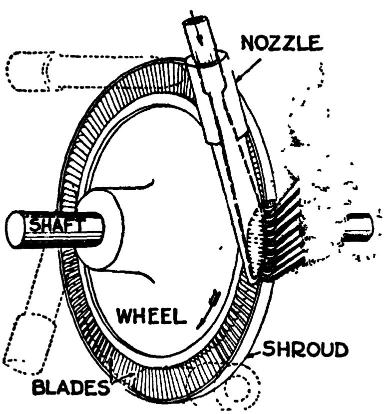


FIG. 140. Diagrammatic turbine, impulse type.

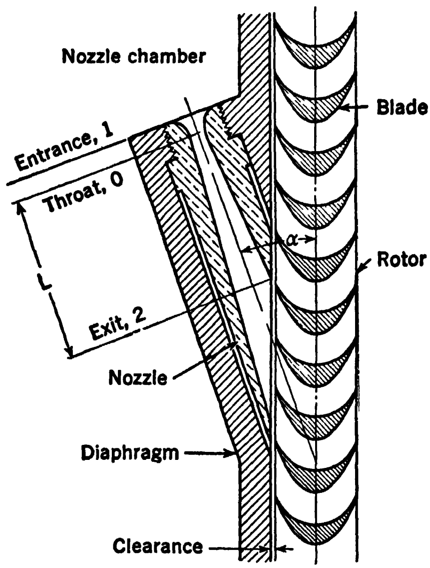


FIG. 141. Typical convergent-divergent steam nozzle for impulse turbine.

206. Steam Flow in Nozzles. The steam in a turbine is continuously directed from nozzles or guide passages against blades uniformly spaced on the periphery of a rotor. The energy transformation is accomplished by forces on the blades of the rotor because of the changes of momentum of the steam in its passage through the blade channels. Thus enthalpy is converted into kinetic energy as the steam flows through the nozzle. In an ideal turbine all of the change in enthalpy of the steam would appear as energy delivered to the shaft. The ideal turbine is, therefore, of value in studying the velocity which a steam jet will acquire, its behavior, and the dimensions of the nozzles necessary.

In Fig. 141, let points 1, 0, and 2 mark the entrance, throat, and exit boundaries, respectively, across which the steam is moving in steady flow. As the steam passes from point 1 to 0 and on to point 2 it is moving from a region of high pressure to one of lower pressure, and as the volume increases each element is accelerated by the expansion of all other elements in the stream. The continuity equation must apply; i.e.,

$$\frac{A_1V_1}{v_1} = \frac{A_0V_0}{v_0} = \frac{A_2V_2}{v_2} = m \quad \text{mass flow} \quad \text{lb per sec} \tag{208}$$

where A = area, sq ft.
V = velocity, fps.
v = specific volume, cu ft per lb.

As the steam flows from point to point the energy conversions are accounted for by the steady-flow equation. If the flow is isentropic, then

$$u_1 - u_2 + \frac{P_1v_1 - P_2v_2}{J} = \frac{V_2^2 - V_1^2}{2gJ} \tag{209}$$

Since $(u_1 + P_1v_1/J)$ is h_1 and $(u_2 + P_2v_2/J)$ is h_2, the above equation may be written as

$$h_1 - h_2 = \frac{V_2^2 - V_1^2}{2gJ} \tag{210}$$

The change of enthalpy is, therefore, equal to the gain in kinetic energy.

In most cases the initial velocity is negligible so that

$$h_1 - h_2 = \frac{V_2^2}{2gJ} \tag{211}$$

where V_2 is the velocity in feet per second, acquired as a consequence of the change in enthalpy. Since $g = 32.174$,

$$V_2^2 = 778.26 \times 64.348(h_1 - h_2)$$

or

$$V_2 = 223.8\sqrt{h_1 - h_2} \qquad (212)$$

207. Nozzle Proportions. In designing an ideal nozzle for constant entropy expansion, the cross-sectional area at any point n may be computed by use of the continuity equation, or

$$A_n = \frac{mv_n}{V_n} \qquad (213)$$

where A_n = area, sq ft.
m = mass of steam flowing, lb per sec.
v_n = net specific volume of steam at pressure P_n, cu ft per lb after isentropic expansion.
V_n = velocity of steam at point n, fps, as determined from equation 212.

Thus the area of the nozzle at any point is a function of the volume, the velocity, and the mass of steam flowing. When the flow is adiabatic and frictionless, the entropy of the steam at any point in the nozzle is equal to the initial entropy. In designing a turbine nozzle it is necessary to choose a throat area so that the desired amount of steam may flow with the desired pressure drop, because the velocity is determined by that pressure drop. This is done by working in terms of 1 lb of steam, since equations, steam tables, and charts give data in terms of 1 lb.

Figure 142 illustrates relations among sectional areas (A), steam velocities (V), and the specific volumes (v) for an ideal nozzle. The graphs indicate that at the beginning of flow in the nozzle, the steam velocity increase is rapid although the corresponding volume increases at a lesser rate. Since mass flow (m) is constant and $V/v = m/A$, it is apparent that A must decrease until the flow has reached that section where the rate of increase in volume is equal to the rate of increase in velocity. At this section V/v is a maximum and A is a minimum. This section is called the *throat* of the nozzle, point 0.

Figure 142 indicates that the upstream section from the throat should be almost as long as the downstream section. This can be avoided with minimum loss by a shortened and well-rounded approach as indicated by the dotted line 0–1. The throat area is the usual starting point in sizing the nozzle. The divergent portion of the nozzle is generally made with a constant angle of divergence of approximately 6 deg from the center line of the nozzle until reaching the required area of exit. Too large an angle of divergence induces turbulence, and if the divergence is too small the nozzle length becomes too great.

The length of the nozzle is not critical and may be proportioned on the basis of the throat area by the empirical relation

$$L = \sqrt{15A_0} \tag{214}$$

where L = length of nozzle from throat to exit, in.
A_0 = area of the throat, sq in.

The cross-sectional shape of nozzles may be either circular, elliptical, square, or rectangular with rounded corners. Experimental work indicates that the efficiency of a nozzle is practically unchanged by the

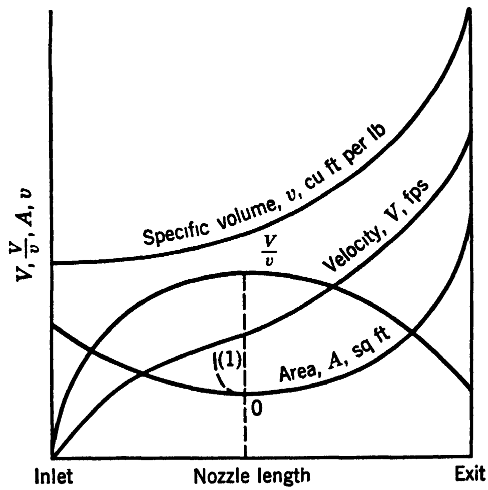


FIG. 142. Nozzle relations.

shape of the cross section when the internal surfaces are smooth, the nozzle proportions have the correct ratio of throat area to exit-end area, and the length of the nozzle is not less than the length calculated by equation 214, nor more than twice that value. In a properly proportioned nozzle having straight sides from throat to exit, *the efficiency as measured by the velocity is about 96 to 97 per cent. This corresponds to an equivalent energy efficiency of 92 to 94 per cent.*

Example. (*a*) Design a theoretically perfect nozzle to pass 1 lb of steam per sec, the steam being initially dry and saturated at a pressure of 150 psia and the final pressure being 14.7 psia. (*b*) If the turbine blades could be made to transform all the issuing jet velocity into useful work, find the horsepower imparted to the rotor. (*c*) What is the theoretical steam consumption (steam rate) in pounds per horsepower per hour?

Solution. (*a*) From steam tables and Mollier diagram, the following values are obtained:

$$p_1 = 150 \text{ lb} \qquad x_1 = 1.0 \qquad s_{g,1} = 1.5694$$
$$p_2 = 14.7 \text{ lb} \qquad x_2 = 0.8704 \qquad s_{g,2} = 1.5694$$
$$p_0 = 87 \text{ lb} \qquad x_0 = 0.9614 \qquad s_{g,0} = 1.5694$$

$$v_{g,1} = 3.015 \text{ cu ft} \qquad h_{g,1} = 1194.1 \text{ Btu per lb}$$
$$v_{g,2} = 23.34 \text{ cu ft} \qquad h_{g,2} = 1024.62 \text{ Btu per lb}$$
$$v_{g,0} = 4.86 \text{ cu ft} \qquad h_{g,0} = 1150.01 \text{ Btu per lb}$$

From equation 212,

$$V_2 = 223.8\sqrt{1194.1 - 1024.62} = 2912 \text{ fps}$$
$$V_0 = 223.8\sqrt{1194.1 - 1150.01} = 1485 \text{ fps}$$

Using equation 213,

$$A_2 = \frac{1 \times 23.34}{2912} \times 144 = 1.154 \text{ sq in.}$$

and

$$A_0 = \frac{1 \times 4.86}{1485} \times 144 = 0.471 \text{ sq in.}$$

From equation 214,

$$L = \sqrt{15 \times 0.471} = 2.66 \text{ in.}$$

The entrance end of the nozzle will be rounded as shown in Fig. 141, the throat diameter is 0.77 in., the diameter of the nozzle exit is 1.21 in., and the length of the nozzle from throat to exit is 2.66 in.

(*b*)
$$\text{KE} = \frac{mV^2}{2g} = \frac{1 \times (2912)^2}{2 \times 32.174} = 131{,}800 \text{ ft-lb per sec}$$

Therefore

$$\text{hp} = \frac{131{,}800}{550} = 239.6$$

(*c*)
$$\text{Steam per hp per hr} = \frac{3600}{239.6} = 15 \text{ lb}$$

The exit end of a turbine nozzle is cut off obliquely, as shown in Figs. 140 and 141, in accordance with the angle made by the nozzle axis with the plane of blade rotation. This inclination is determined as a compromise between conflicting considerations. Higher theoretical efficiency can be obtained by using small nozzle entrance angles because the blade action will be more efficient. Under these conditions a long nozzle is required, and the frictional loss is greatly increased; the total deviation of the steam in the blading is increased, causing ad-

ditional frictional losses; and the diagonally cut exit end of the nozzle is so wide that it fails to direct the steam into the blading properly. If this nozzle angle becomes too great, the blading loss becomes large because the steam jet strikes the back of the blades at the entrance. Nozzle entrance angles found in practice range from 10 to 30 deg, 15 to 20 deg being common.

208. Critical Pressure in Nozzles. If the mass flow (pounds per second) is plotted against the ratio of exit to inlet pressure for a given nozzle in which the inlet, throat, and exit areas (A_1, A_0, A_2) are fixed, a graph of the type shown in Fig. 143 is obtained. When $P_2/P_1 = 1$, it is evident that there will be no flow. As the exit pressure becomes less than the inlet pressure, flow increases along b to c. With further decrease in exit pressure, the flow does not increase, as might be expected, but remains constant along c to a.

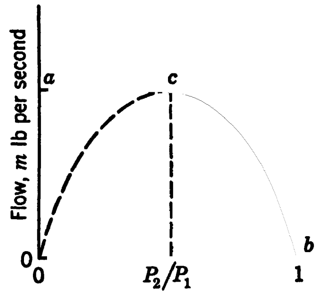


FIG. 143. Flow characteristics of a nozzle.

The point c, where $P_2/P_1 = 0.58$, is called the *critical pressure* (P_c) for wet steam. The critical pressure is equal to 0.54 for superheated steam. For air and other gases for which $k = 1.4$, the critical pressure is 0.53. These values are rational and may be computed for gases for which the value of k is known.

When the critical pressure has been reached, the medium flows at the velocity of sound, and, since no disturbance in the medium is propagated in excess of this velocity, a further decrease in pressure at the exit has no effect on the pressure above the throat. Consequently, the flow is at a maximum and is constant for all values of exit pressure less than the critical. Above the critical pressure the flow is a function of the exhaust pressure.

209. Deviations from Isentropic Conditions. As a first approximation it is customary in considering the behavior of a nozzle, or of a turbine, to assume the process to be isentropic. The deviation from the ideal can be evaluated by coefficients which account for the difference between the actual and the ideal.

Thus, the coefficient of discharge, n_d, for a nozzle is defined as

$$n_d = \frac{\text{Actual mass flow}}{\text{Ideal mass flow}} \tag{215}$$

For well-designed nozzles n_d is close to 0.98 for gases and vapors.

The term *nozzle coefficient*, n_n, is defined as

$$n_n = \frac{V_{2_{actual}}}{V_{2_{isentropic}}} \quad \text{or} \quad V_{2_{actual}} = 223.8 n_n \sqrt{h_1 - h_2} \qquad (216)$$

where h_1 = the enthalpy before isentropic expansion, Btu per lb.
h_2 = the enthalpy after isentropic expansion, Btu per lb.

Values of n_n range between 0.94 and 0.98 for steam as long as the moisture does not exceed 10 per cent.

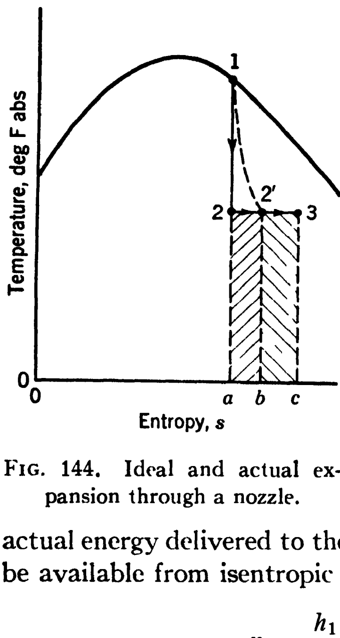


FIG. 144. Ideal and actual expansion through a nozzle.

Figure 144 illustrates the effect of irreversibility in the nozzle and turbine blades. The crosshatched area *a–2–2′–b–a* shows the increase in unavailable energy due to friction in the nozzle. As the steam strikes the blade, friction, shock, and turbulence result in a further increase in unavailable energy as indicated by the area *b–2′–3–c–b*. The ideal case assumes that steam would expand from point 1 to point 2 at constant entropy, and all the resulting kinetic energy would be converted perfectly to shaft work by the turbine blades, the exit steam remaining static at point 2.

Stage efficiency (n_s) is defined as the actual energy delivered to the blades, divided by the energy that should be available from isentropic expansion.

$$n_s = \frac{h_1 - h_3}{h_1 - h_2} \quad \text{(See Fig. 144)} \qquad (217)$$

If the successive state points for steam, as it flows from stage to stage through a turbine, are plotted on a Mollier diagram, the result is called a *condition curve*. In an actual turbine the condition curve can be obtained experimentally by taking a pressure and a temperature measurement between successive stages if the steam is in the superheated region. If it is in the saturated region the steam quality must also be known and a calorimeter reading is necessary. Such a curve is shown in Fig. 145.

210. Impulse Turbines. Figure 146 illustrates the impulse turbine in which all the pressure drop has occurred in the nozzle and a part of the resulting kinetic energy of the flowing steam has been absorbed by the moving blades. The nozzles of impulse turbines may not extend

completely around the periphery of the turbine; therefore at a particular time only part of the rotor blades can be acted upon by the steam jets. The earliest turbines of this type were the De Laval "Class A" units, the essential elements of which are shown in Fig. 140. The steam was expanded from the initial to the exhaust pressure in the nozzles, and the kinetic energy of the jet was absorbed by a single row of blades fastened to the periphery of the wheel. The blades deflect

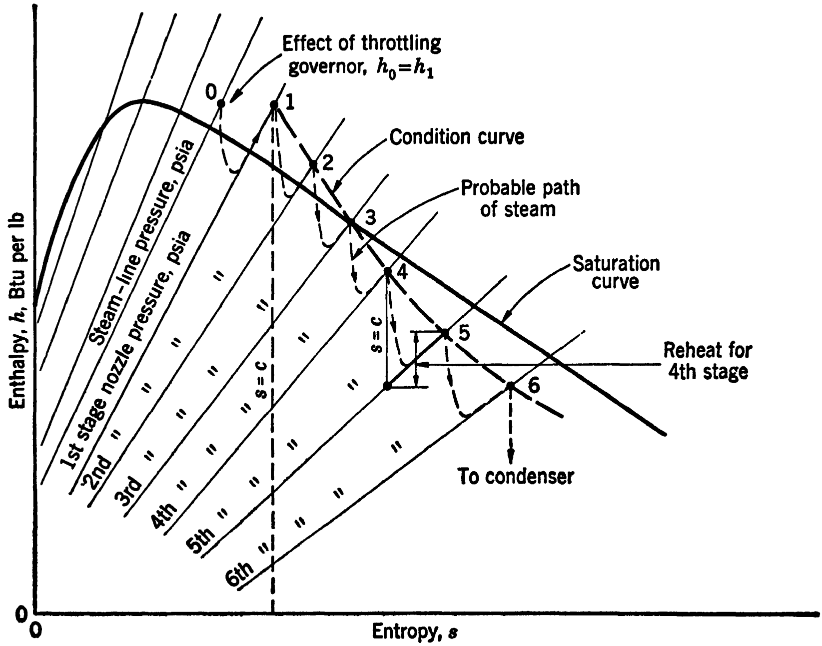


FIG. 145. Condition curve for a steam turbine.

the steam jet as shown and absorb the greater part of the kinetic energy so that the steam flows away from the wheel with a low velocity. Some turbines of this type are in use, but they are no longer manufactured because of their high rotative speeds, 10,000 to 30,000 rpm.

In an ideal impulse turbine the steam jet issuing from a nozzle would be brought to rest in the blades and would thus give up all its kinetic energy. In the actual turbine it is impossible to bring the jet to rest because of the limitations of practical designs. There is, therefore, always an energy loss in a turbine because of the residual or terminal velocity of the steam leaving the wheel.

Figure 147 shows a section of a blade in a single-stage impulse tur-

bine. If the minimum eddy loss at entrance is to be obtained, the jet must enter along a tangent to the inside curve at the entrance edge. This direction is shown by the line V_r in the figure, and, if the blade were stationary, the steam jet would move as shown by V_r. Since the blade moves ahead and away from the jet, the steam must be directed

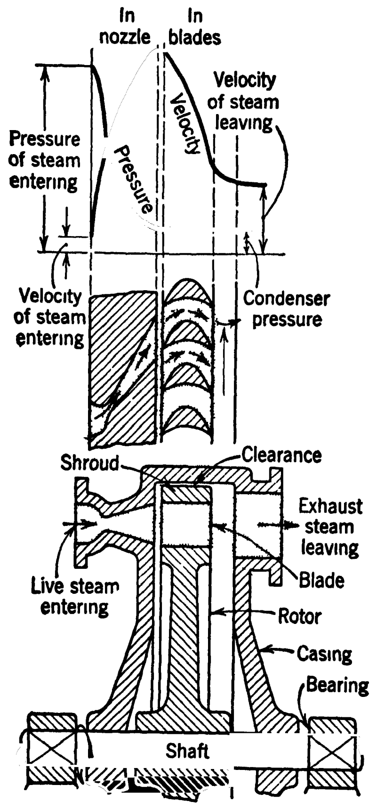


FIG. 146. Impulse turbine, single-pressure single-velocity stage.

as indicated by V_a in order to strike the blade in the direction indicated by V_r.

In the case of steam impinging on a blade, as shown in Fig. 147, the nozzle directs the jet against the blade with absolute velocity V_a. A nozzle angle α (about 20 deg with the plane of the wheel) is used. Since the blade moves at a velocity of V_b, the velocity V_r of the steam jet relative to the blade is the geometrical resultant of V_a and V_b. For convenience in construction, the entrance angle β and exit angle β' are

made equal. Neglecting frictional losses in the blade, the relative exit velocity V'_r equals V_r, and the resultant of V'_r and V_b is the absolute exit velocity V'_a. Obviously, the energy removed from steam passing over the blade is

$$\frac{mV_a^2}{2g} - \frac{m(V'_a)^2}{2g} \tag{218}$$

where m is the mass flow of steam, lb per sec.

If the blade were theoretically perfect, the absolute velocity V'_a would be zero and all the kinetic energy would then be removed. This

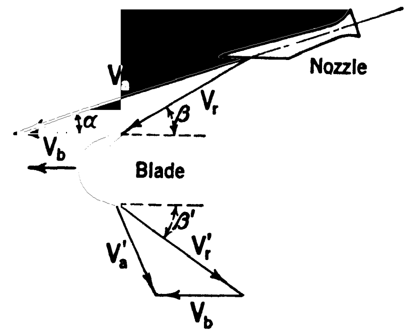


FIG. 147. Velocity diagram for impulse turbine blading.

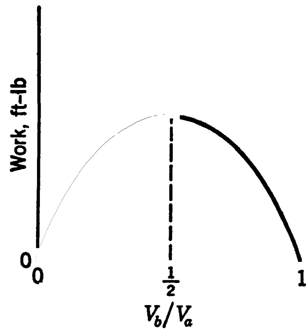


FIG. 148. Shaft work vs. blade- and steam-velocity ratios, for impulse turbines.

is practically impossible in an actual turbine, and there is always a loss due to the residual velocity V'_a. The best that can be done is to choose the nozzle angle α and the velocity of the blade with respect to the steam jet velocity so that the value of V'_a will be a minimum. For the production of maximum work from a single-stage impulse turbine the blade velocity should be about one half of the absolute velocity of the steam jet. This statement is illustrated in Fig. 148, which shows that at stalled conditions ($V_b/V_a = 0$) no work is done, because, although the torque on the blades may be maximum, the distance through which the torque acts is zero. Likewise when the speed of the blade is equal to that of the steam jet the torque will be zero although the distance traveled by the blade will be maximum and no work will be done.

211. Steam-Turbine Staging. Turbine staging has the purpose of decreasing the wheel speed and still keeping the blade velocity near the optimum with respect to the jet velocity, i.e., essentially one-half jet velocity for a single-stage impulse wheel and the equivalent of the jet velocity for the reaction wheel. The velocity of a steam jet may be

very high, depending on the initial pressure and temperature of the steam as well as the back pressure. If all the energy were converted in one stage, a turbine speed of 20,000 to 40,000 rpm would be necessary. Such speeds would require reduction gearing of undue proportions.

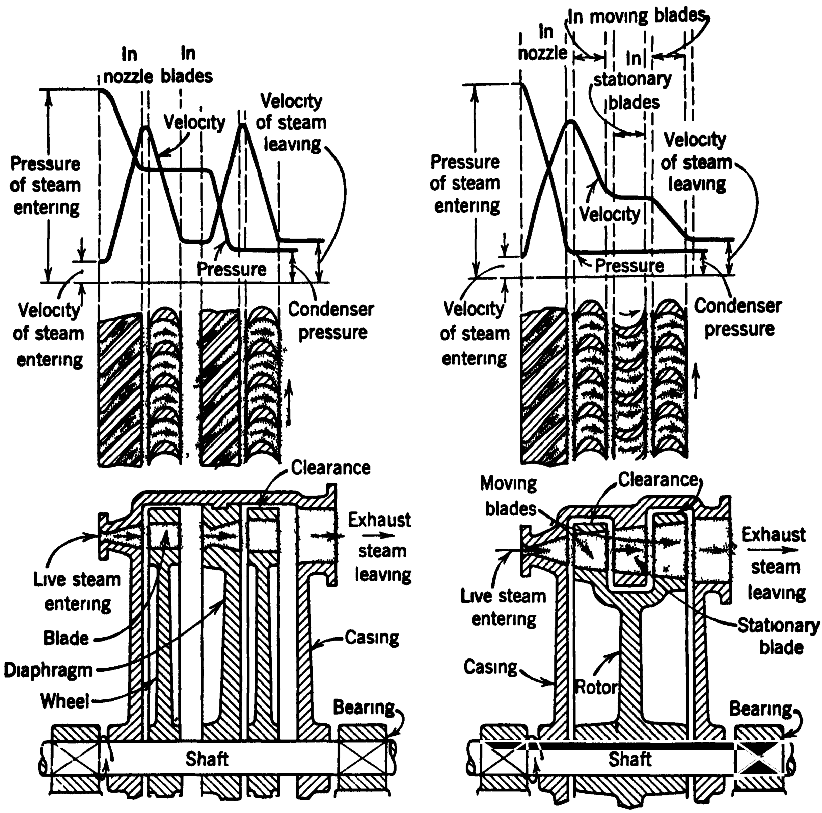


FIG 149. Impulse turbines (*a*) Rateau or pressure staging and (*b*) Curtis or velocity staging. Note pressure and velocity graphs.

The two types of staging commonly used in impulse turbines are (1) *pressure* and (2) *velocity*. In pressure staging, the drop in pressure is limited in any one set of nozzles so that the resulting velocity is low enough to be absorbed by a reasonable wheel speed. This process is repeated as many times as necessary to expand the steam fully and is commonly called the *Rateau principle of staging.*

Velocity staging allows a large pressure drop in one set of nozzles, and the resulting steam velocity is then utilized by as many sets of

rotor blades as are necessary. This method of staging is known as the *Curtis principle.* Figure 149(*a*) illustrates a two-pressure-stage (Rateau) impulse turbine, and Fig. 149(*b*), a two-velocity-stage (Curtis) impulse turbine. Figure 146 shows a single-pressure single-velocity-stage impulse turbine. In each figure there is a graph representing the pressure-velocity relations for each turbine.

A turbine employing both velocity and pressure staging is illustrated by Fig. 150. The velocity developed in the first-stage nozzles C is utilized in two rows of moving blades D and F. Blades D absorb part of the velocity; the steam is next reversed by stationary blades E, and directed into the second row of moving blades F, which absorb most of the remainder. The steam is then expanded in the entrance nozzles of nine successive pressure stages, each of which has one row of moving blades, i.e., one velocity stage per pressure stage. Each set of nozzles converts a portion of the available energy, which when transformed into velocity results in a steam-jet speed of about twice the blade speed.

The energy-transforming capacity of the Curtis stage is greater than that of the Rateau, with fewer stages and cheaper construction, but the Rateau principle is more efficient. All impulse turbines designed for high efficiency use either Rateau staging following a Curtis stage or Rateau staging throughout. The turbine shown in Fig. 150 offers the advantages of a large pressure drop in the first-stage nozzles C, whereby the steam enters the casing of the turbine at a lower pressure than if the first stage had been of the Rateau type.

Example. A steam turbine of the impulse type is to be used for driving a generator at 3600 rpm; the mean diameter of the wheels is to be 48 in. The quantity of dry steam to be supplied is 10,800 lb per hr at a pressure of 200 psia. Assuming isentropic expansion to a condenser pressure of 2 in. of mercury abs, and neglecting friction, find (*a*) the theoretical number of velocity stages required, (*b*) the theoretical number of pressure stages, and (*c*) the number of pressure stages if there are two velocity stages for the first pressure stage.

Solution. (*a*) *Velocity staging.*

$$\text{Peripheral velocity} = \tfrac{3600}{60} \times \tfrac{48}{12}\pi = 754 \text{ fps}$$

$$\text{Mass flow} = \frac{10{,}800}{60 \times 60} = 3 \text{ lb per sec}$$

$$1198.4 - 862.1 = 336.3 \text{ Btu per lb available}$$

$$\text{KE} = \frac{mV^2}{2g}$$

$$336.3 \times 778.26 = \frac{V^2}{64.34}$$

$$V = 4102 \text{ fps}$$

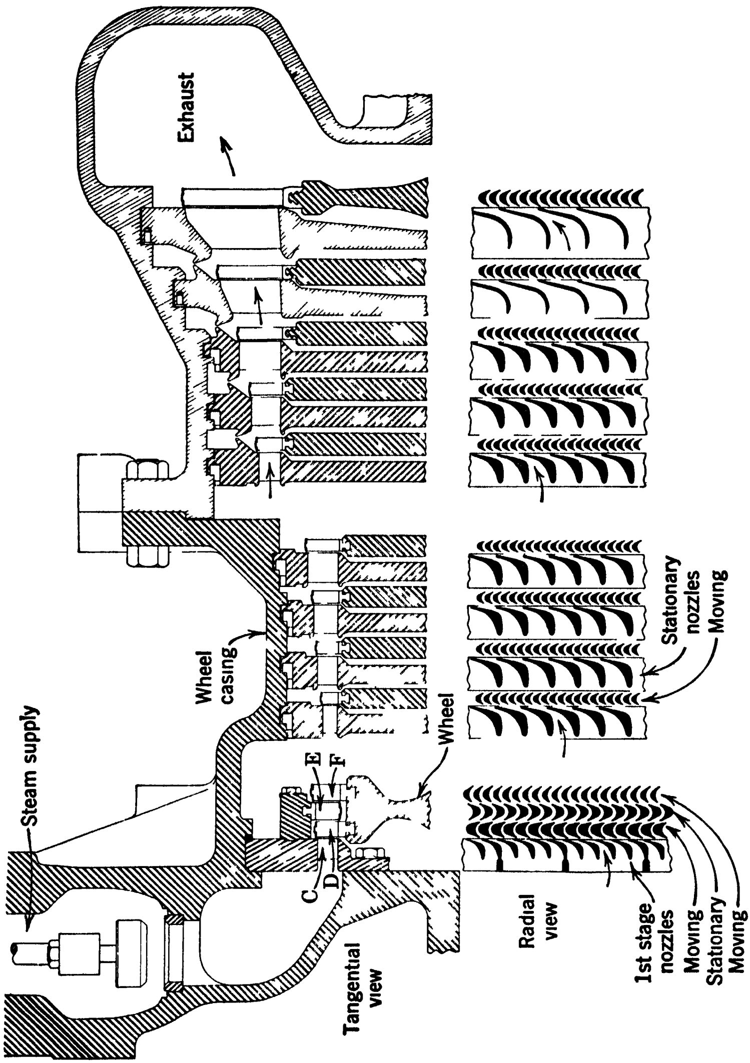


FIG 150 Multistage impulse steam turbine with ten pressure stages, the first pressure stage has two velocity stages, the other nine have one velocity stage each

Ideal steam velocity per stage = 2 × 754 = 1508 fps; thus

$$N_v = \frac{4102}{1508} = 2.72$$

Use two velocity stages.

(*b*) *Pressure staging.* Steam velocity to be absorbed per stage = 1508 fps. Therefore for 1 lb of steam

$$KE = \frac{mV^2}{2g} = \frac{1 \times (1508)^2}{64.34} \text{ ft-lb per stage}$$

$$KE = 35{,}344.3 \div 778.26 = 45.43 \text{ Btu per stage}$$

But 336.3 Btu are available; thus

$$N_p = \frac{336.3}{45.43} = 7.4$$

Use seven pressure stages.

(*c*) *One Curtis stage (double wheel) and Rateau staging.* Steam velocity required for one double-row Curtis stage = 2 × 2 × 754 = 3016 fps.

$$KE = \frac{mV^2}{2g} = \frac{1 \times (3016)^2}{64.34} \text{ ft-lb per stage}$$

$$KE = 141{,}378 \div 778.26 = 181.7 \text{ Btu for double-row Curtis rotor}$$

For Rateau stages, 336.3 − 181.7 = 154.6 Btu are available. From part (*b*), each Rateau stage absorbs 45.43 Btu per lb; thus

$$N_r = \frac{154.6}{45.43} = 3.4 \text{ Rateau stages}$$

The turbine would consist of one double-row Curtis and three Rateau stages.

In making a comparison of the results obtained in the preceding example, it might seem that velocity staging is better because a smaller number of stages is required. As stated previously, the advantage of the smaller number of stages is overbalanced by the fact that pressure staging is more efficient. Turbines with pressure staging are used extensively and are often called multicellular because each pressure stage is made up of a compartment that is steamtight except for the openings through the nozzles. However, where no difference in pressure exists, as in velocity staging, there is no leakage. In the velocity-stage type, the pressure beyond the nozzles is the same throughout the entire turbine casing and thus there is less interstage leakage. Curtis rotors of three velocity stages may be used, but generally there are not more than two because the high-velocity steam jet is broken up by its passage through the blades, and the efficiency is low after the first two stages. A very high entering steam velocity also produces "skin" friction between the steam and the blade surfaces, causing a considerable loss.

212. Single-Pressure Multivelocity-Stage Impulse Turbines. Figure 151 shows a longitudinal section through a single-pressure two-velocity-stage noncondensing steam turbine embodying the Curtis principle. The rotor consists of a shaft, a double-row wheel, and a speed governor. The revolving blades and stationary guide vanes are made of stainless steel and are inserted in dovetail slots in the wheels

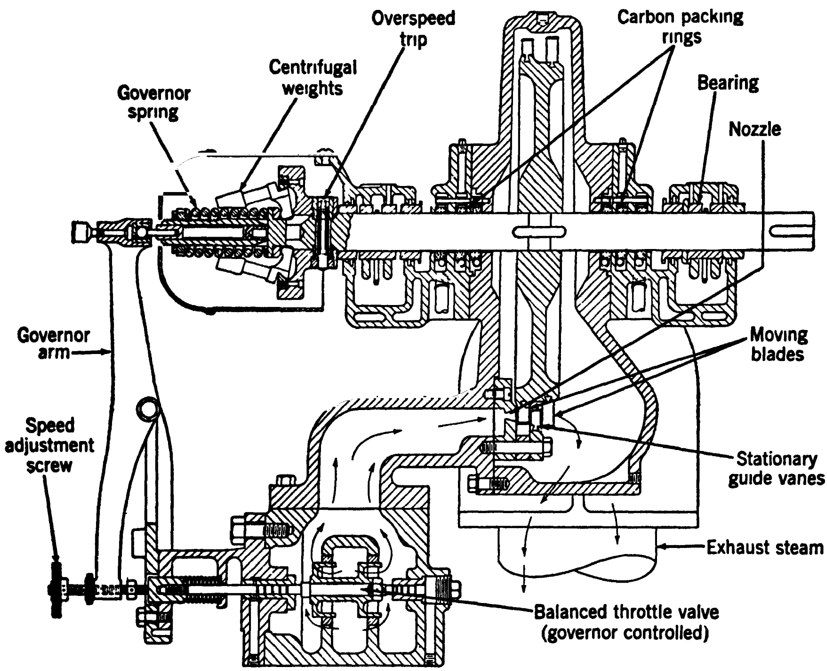


FIG. 151. Longitudinal section of two-velocity-stage steam turbine.

and stationary holder. The nozzles are of the diverging type and are machined in a nozzle ring casting which is bolted to the side of the steam ring. The spacing is very close so that the steam jets unite upon leaving the nozzles, and the blades, therefore, receive a continuous band of steam. The intermediate stationary guide vanes receive the steam from the first row of blades and direct the steam upon the second row of blades. As complete expansion takes place in the nozzles, uniform pressure exists throughout the remainder of the turbine; hence there is no tendency for leakage of steam over the tops of the blades. This fact permits the use of large running clearances. Steam leakage along the shaft is partly prevented by carbon packing rings.

The speed of the turbine is controlled by a centrifugal governor

mounted on the end of the main shaft and attached, through linkages, to a balanced throttle valve in the steam supply line. Two governor weights are mounted upon hardened-steel knife-edges in a cage that rotates. A coil spring resting against a sliding sleeve loads the governor so that any change in the spring compression varies the speed of the turbine.

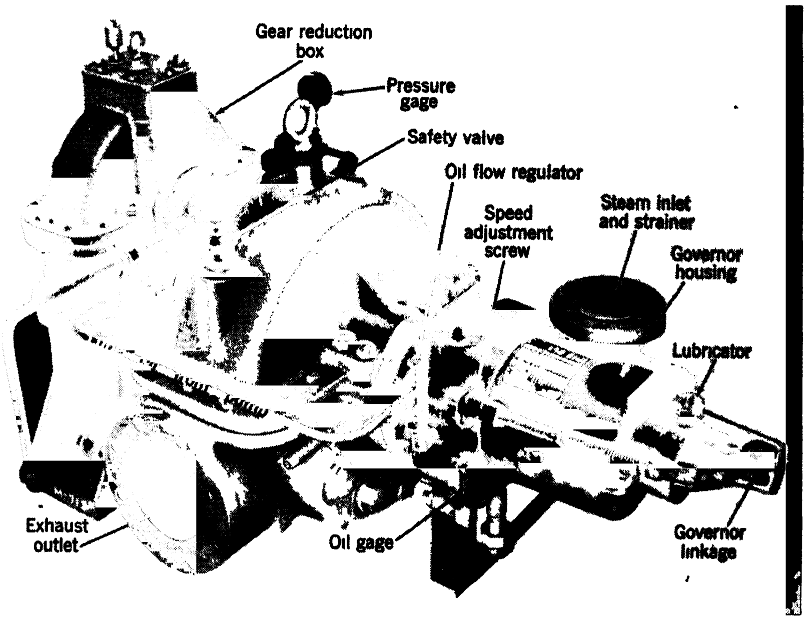


FIG 152 Worthington noncondensing turbine.

Machines suitable for high speeds, such as high-head centrifugal pumps, centrifugal blowers and compressors, and electric generators, can be directly connected through a flexible coupling to steam turbines of this class Reduction gears are used between the turbine and any machine restricted to speeds lower than those of the turbine, such as large centrifugal pumps and blowers, belt pulleys, and paper mill and other moderate-speed machinery. Turbines of this type are available in sizes up to 3000 hp and are primarily designed for noncondensing service.

Figure 152 is an exterior view of a geared unit having a 16-in. diameter wheel and equipped with a direct-acting throttle-type governor. Figure 153 is a sectional view of the same unit. The casing is designed to accommodate either a two-stage or a three-stage Curtis wheel. In

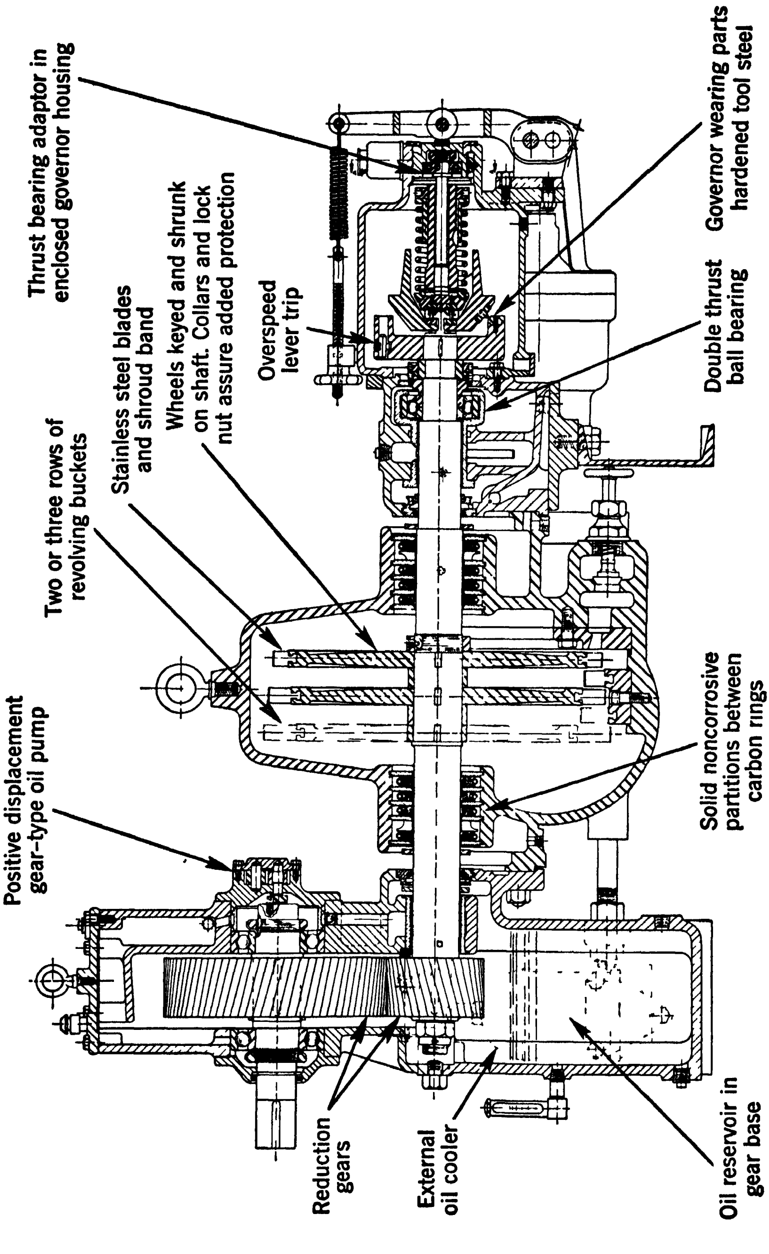


FIG. 153. Longitudinal section of Worthington multivelocity-stage turbine with reduction gear.

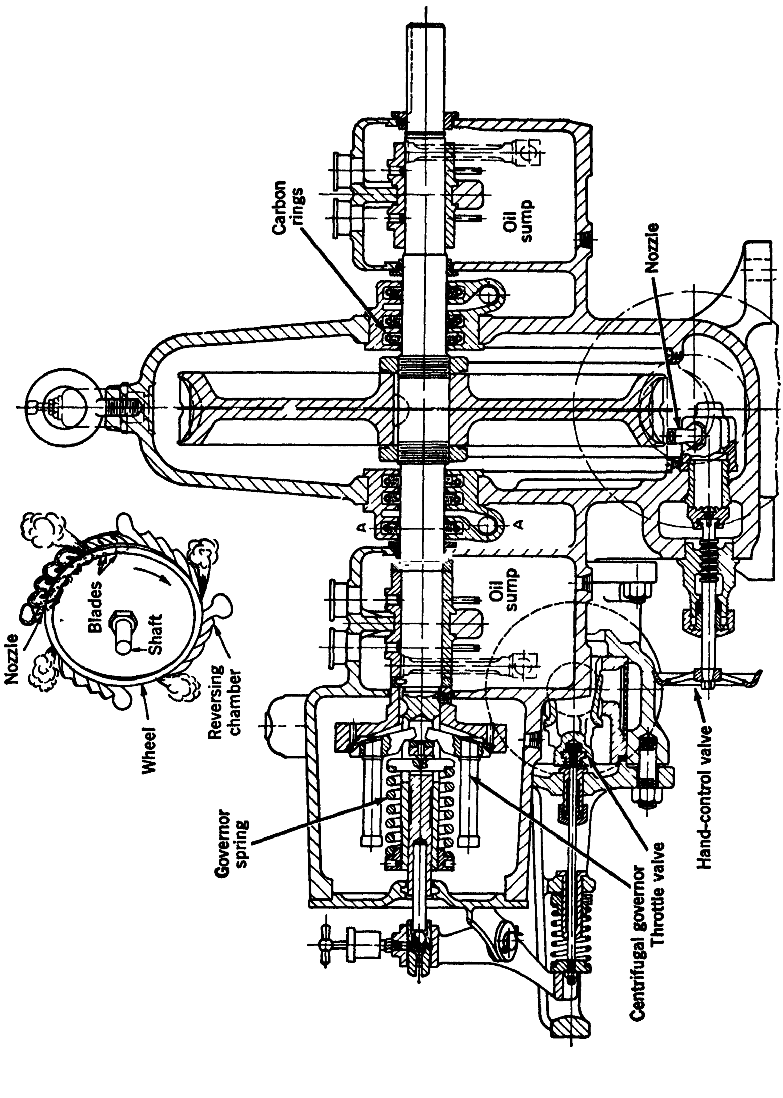


FIG. 154. Terry re-entrant turbine.

power plant applications it would operate noncondensing with the exhaust heating the feedwater.

The impulse turbine shown by Fig. 154 is known as a *re-entrant* type because the steam re-enters the single rotor after leaving the nozzle. This is accomplished by locating the nozzles directly in line with the edge of the rotor. Steam issuing from the nozzle flows into cup-shaped recesses in the wheel and is reversed 180 deg. As the steam emerges it is redirected for entrance again into the rotor by spiral-shaped cover plates as illustrated by the insert, Fig. 154. This process is repeated until the initial velocity developed in the nozzle is absorbed and the

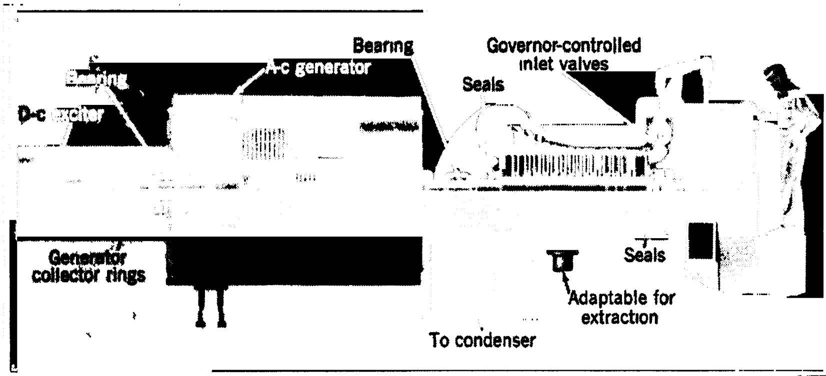


FIG. 155. Allis-Chalmers WA-series impulse steam turbine.

steam floats out through openings in the cover plate. The steam nozzles are generally equipped with hand-control valves which permit the use of the required number of nozzles for best efficiency at various loads. Turbines of this type are usually used for driving auxiliaries and are operated noncondensing, the exhaust being used for process work or heating.

213. Single-Cylinder Condensing Impulse Turbine. Figure 155 shows a cutaway view of a condensing impulse turbine built in sizes from 2000 to 7500 kw. The unit is also available for noncondensing and automatic extraction-condensing applications. The first stage consists of a two-velocity Curtis wheel, and the remaining stages are single pressure single velocity.

Multiple governing valves, hydraulically controlled, minimize throttling losses and regulate the steam flow according to load demands. As a safety feature, an independent overspeed governor is provided to shut off the steam in case the normal operating speed should be exceeded by 10 per cent. An additional safety feature prevents steam from flowing

to the turbine unless the lubricating oil pressure is adequate. Metallic labyrinth packing is used to provide steam seals at the glands. The turbine buckets and nozzles are made of chrome alloy steel, machined from solid bar stock. The blades are held in T-slots in the wheels and are equipped with shroud bands to minimize leakage at the blade tips and to prevent blade vibration. The turbine wheels and shaft are individually balanced after machining. The wheels are then shrunk and keyed to the shaft, after which the rotor is dynamically balanced. The lubrication system consists of a main oil pump, an auxiliary pump, a cooler, and an oil reservoir. Both pumps are gear type and submerged in the reservoir. One is driven from the turbine rotor; the other is steam-turbine driven with automatic controls to maintain a predetermined oil pressure.

A typical single-cylinder condensing impulse turbine designed for two-stage extraction is illustrated in Fig. 156 with an index of parts.

214. Reaction Turbines. In the theoretical reaction turbine the expansion of the steam occurs in the moving blades. Turbines commonly designated as reaction turbines employ both the impulse and the reaction principles. This type of reaction turbine is shown diagrammatically in Fig. 157. The steam enters from the left and passes through a set of stationary blades (stator) in which occurs some drop in pressure. In passing through the next set of blades, which are moving, a further drop in pressure occurs. The velocity acquired in the stationary blades and also that produced in the rotor blades is absorbed by the rotor blades. The jet with low residual velocity is deflected to the second set of stationary blades with another drop in pressure, seldom exceeding 3 lb, and then enters the second rotor at practically the same velocity as that of the blades. Considerable expansion takes place in the moving blades, and thus, because the pressure is not the same on both sides of these blades, the turbine is called a *reaction turbine.* A stage of a reaction turbine consists of one stationary element and one moving element.

Example. A steam turbine of the reaction type is to be used for driving a generator at 3600 rpm; the mean diameter of the rotor is to be 48 in. Assuming isentropic expansion and neglecting friction, find the number of stages required in a "theoretical" reaction turbine if 10,800 lb of dry steam are supplied per hr at a pressure of 200 psia, the condenser pressure to be 2 in. of mercury absolute.

Solution. *In order to obtain the highest efficiency, the velocity of the moving blades of a reaction turbine should be approximately the same as that of the steam.*

Blade velocity, $V_b = \frac{3600}{60} \times \frac{48}{12}\pi = 754$ fps; therefore the steam velocity $= 754$ fps.

Consider 1 lb of steam since the staging is independent of mass flow.

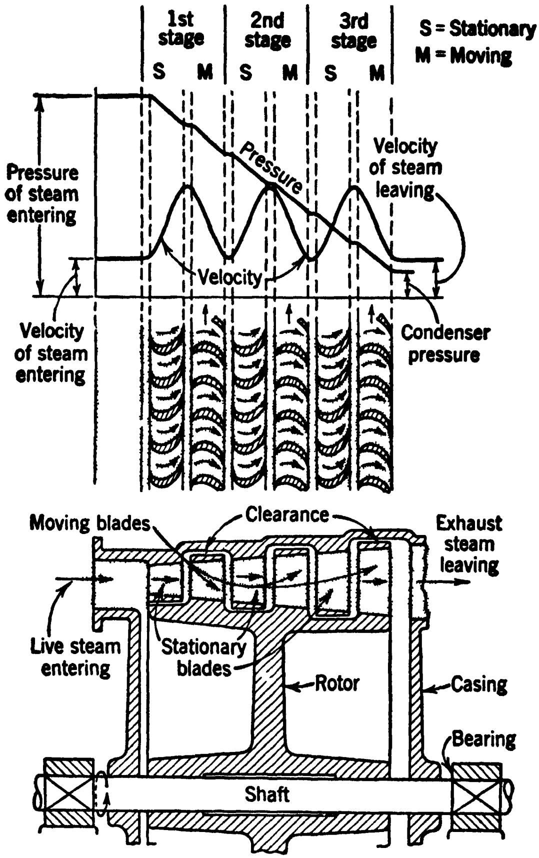


FIG. 157. Reaction turbine, three stages. Note pressure and velocity graphs.

Therefore

$$KE = \frac{mV^2}{2gJ} = \frac{1 \times (754)^2}{2 \times 32.17} \times \frac{1}{778\ 26}$$

$$KE = 11.37 \text{ Btu enthalpy drop per half-stage per second}$$

$$\text{Total enthalpy drop} = (1198.4 - 862.1) = 336.3 \text{ Btu per lb}$$

$$\text{Theoretical reaction stages} = \frac{336.3}{11.37(2)} = 14.8$$

The preceding example shows that the reaction turbine requires more stages than the impulse-type turbine. The axial-flow reaction turbine is often called the Parsons turbine after its inventor, Sir Charles A. Parsons. In the radial-flow reaction turbine, the steam flows outward from

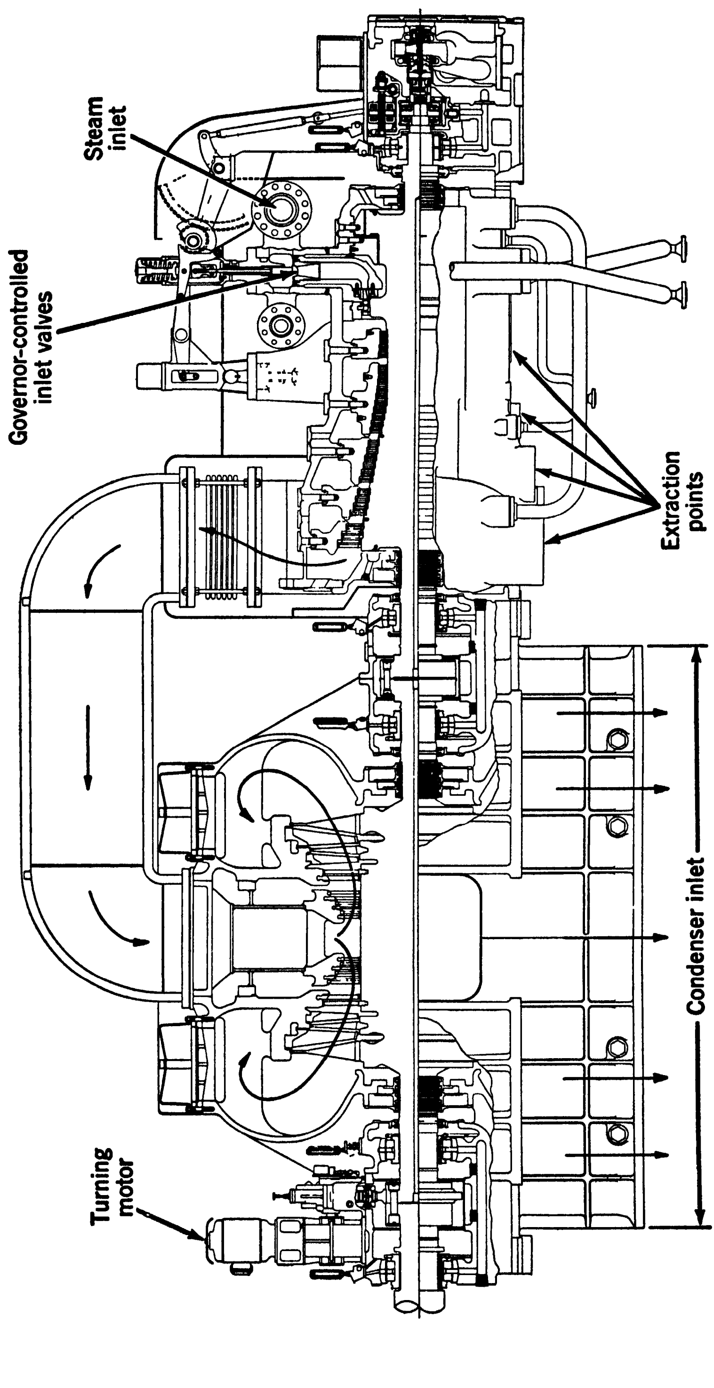


FIG. 158. Allis-Chalmers 66,000-kw tandem-compound turbine.

the shaft to the periphery of the rotor, in this way accommodating the increased volume due to expansion. The Ljungström (Swedish) is the best known of the radial-flow reaction turbines.

215. Two-Cylinder Tandem-Compound Turbine. The tandem-compound turbine shown in Fig. 158 is a cross section of a 66,000-kw, 3600-rpm unit, which is designed to use steam at 1250 psig and 950 F with exhaust at 1.5 in. mercury absolute. It has a normal overload

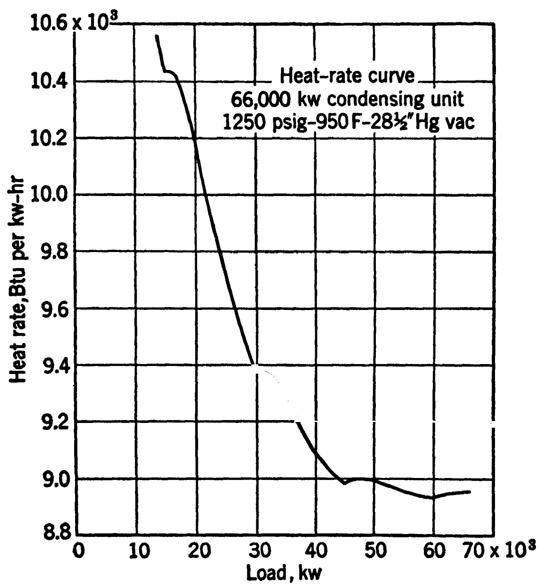


FIG. 158*a*. Heat-rate curve for turbine shown in Fig. 158.

rating of 10 per cent when exhausting against pressures as high as 2½ in. mercury absolute and extracting for normal feedwater heating, the steam requirement being 561,000 lb of steam per hr. Five-stage extraction is provided with steam conditions as follows: 276.7 psia and 410 F; 134.6 psia and 350 F; 53.2 psia and 285 F; 22.8 psia and 235 F; 6.72 psia and 175 F. The normal steam consumption for design conditions is 7.43 lb per kw-hr at 50 per cent load, 7.19 lb per kw-hr at 100 per cent load, and 7.25 lb per kw-hr at 110 per cent load. The generator is hydrogen cooled, and the pressure may be varied by supercharging as high as 15 psig for maximum output. The turbine uses a two-velocity-stage Curtis wheel as the first stage, the remainder being single-pressure single-velocity stages. The heat rate as a nonextraction condensing unit is shown in the graph of Fig. 158*a*. Expansion

problems at high pressures and high temperatures require separate nozzle chests and individually controlled valves, as shown in Fig. 159.

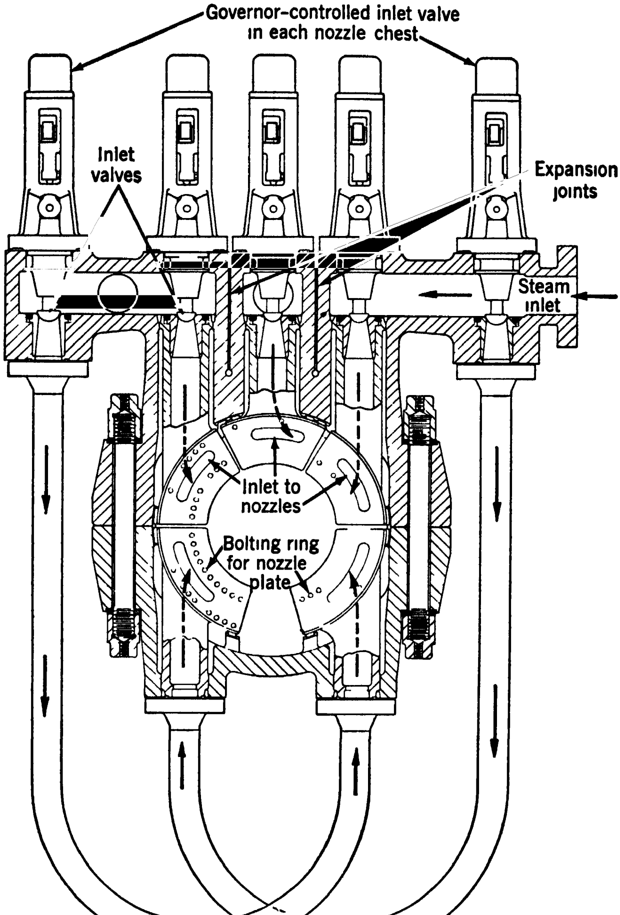


FIG. 159. Allis-Chalmers arrangement of separate nozzle chests (1250 psig and 950 F maximum).

216. Back-Pressure, Extraction, and Low-Pressure Turbines. The name *back-pressure turbine* is given to a noncondensing unit which exhausts to another steam user, operating at a lower pressure. This type can be used to increase the capacity of an existing plant. In such an arrangement the back-pressure unit is called a *topping* or *superposed turbine.* The exhaust steam from the superposed turbine flows to some

or all of the previously installed low-pressure turbines. This arrangement increases the capacity of a given plant and makes possible the economies of high-pressure operation.

Turbines that allow steam to be "bled" at different points are called *extraction turbines*. Figure 156 shows a turbine designed for extraction

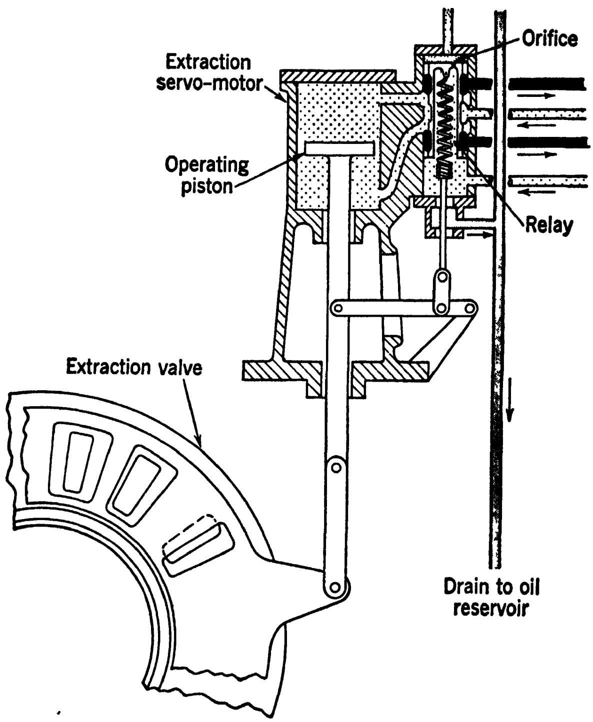


FIG. 160. Grid valve for automatic extraction of steam.

at two points. In many plants, steam at low pressures is needed for process work such as heating, drying, and distilling. For such low-temperature uses, steam may be partially expanded through a turbine, thereby generating as much power as it is possible to obtain before extracting it for process purposes. Such an arrangement has a great thermodynamic advantage for the over-all system, since the process heater serves as a condenser for the turbine. Instead of discarding the energy of condensation to the cooling water, it is utilized in the process. The condensate may be recovered for boiler feedwater as in a surface condenser. The automatic extraction is usually controlled by a grid-type valve as illustrated in Fig. 160. The steam consumption for an

extraction turbine is shown in Fig. 161 for different rates of extraction. The unit steam rate at any load for any amount of extraction and the enthalpy of the "bled" steam may be calculated from the diagram.

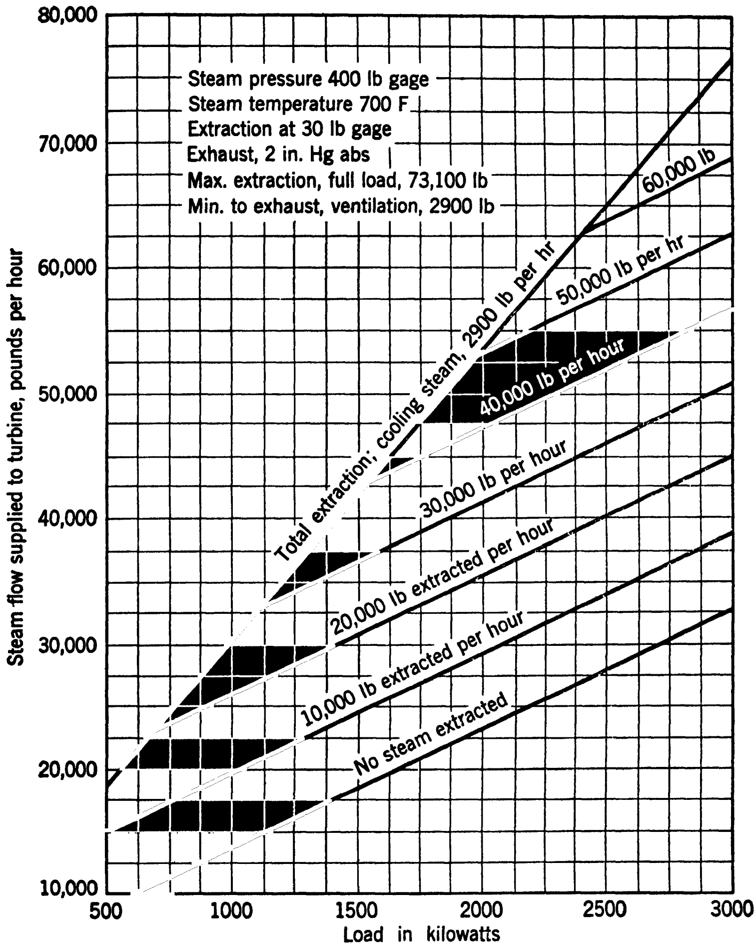
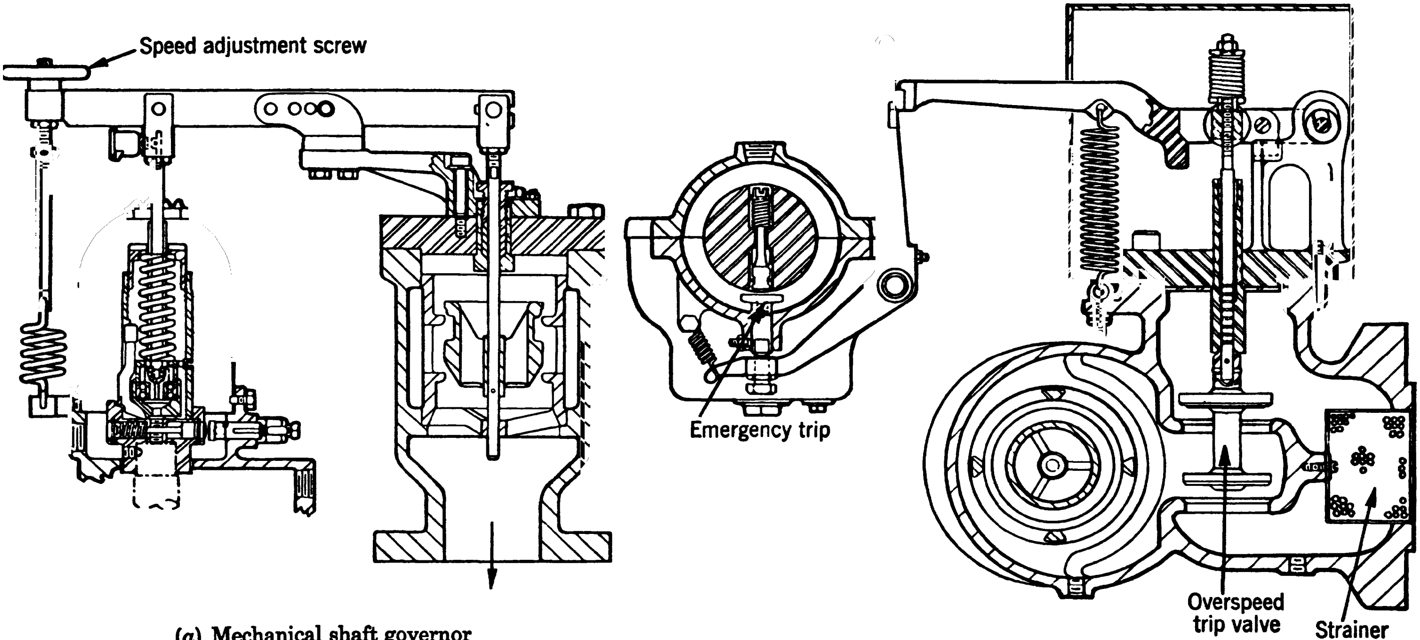


FIG. 161. Steam consumption curves of 3000-kw single automatic extraction steam turbine; 3600 rpm.

Just as it is possible to extract steam from a turbine, it is also possible to add live steam and complete its expansion in the turbine. To prevent excessive moisture concentration in the last stages, dry or superheated steam may be supplied downstream from the main inlet point. This practice depends on the initial quality of the steam, the amount



(*a*) Mechanical shaft governor and throttle valve

(*b*) Overspeed governor and valve

FIG. 162. Elliott constant-speed throttling-type governor and overspeed trip.

extracted, the turbine load, and the exhaust pressure. Approximately 15 per cent moisture in the steam is considered the maximum amount permissible. High moisture concentration will cause excessive blade erosion.

Since turbines can handle large quantities of steam at low pressures with high efficiency, so-called *low-pressure turbines* are frequently operated on inlet steam at atmospheric pressure, the steam then being expanded to 1 or 2 in. mercury absolute pressure in the condenser.

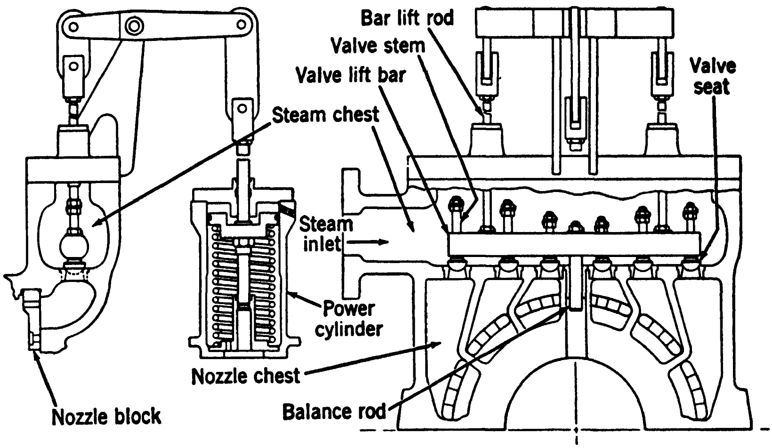


FIG. 163. Allis-Chalmers type of governing valves and operating mechanism.

217. Steam-Turbine Governing. Steam turbines are governed by either *throttle* or *nozzle controls*. Throttle control, Fig. 162(*a*), is the most widely used, particularly on small turbines, because its initial cost is less and the mechanism is simple. The flow of steam entering the turbine is restricted by a balanced valve that is controlled by centrifugal governor weights. As these weights move out radially with increased turbine speed, a linkage adjusts the valve setting to decrease the steam flow. The spring tension opposing the centrifugal weights can be changed by a hand screw to adjust the speed range. Figure 162(*b*) shows a typical overspeed governor.

Nozzle control is accomplished as shown in Fig. 163. Poppet-type valves uncover as many steam passages as necessary to meet the load, each passage serving a bank of nozzles. Figure 164 shows another method for automatic control of inlet valves. A power piston moves the gear segment which in turn moves the cams, and pivoted cam levers open and close valves according to load demands.

The nozzle-type governor has the advantage of using steam at essentially full boiler pressure, and throttling occurs only in that valve which

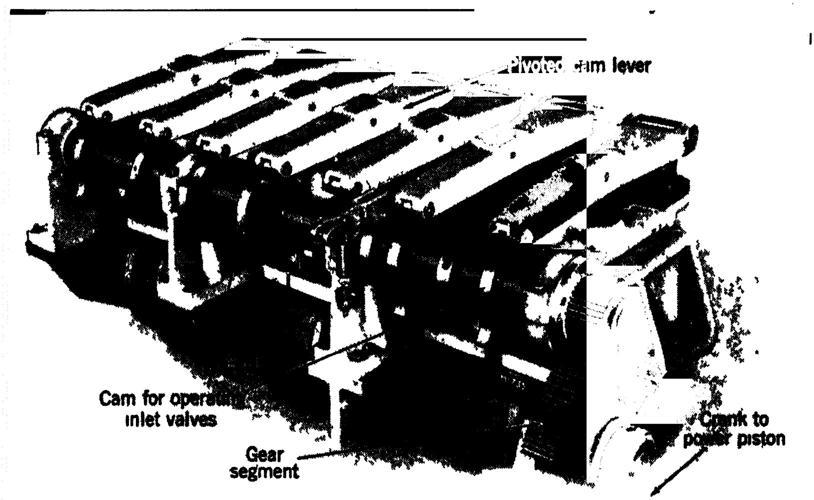


FIG 164 Worthington automatic nozzle control valve.

is partly open; whereas, in throttle governing, all the steam is reduced in pressure before reaching the turbine when the unit is at part load.

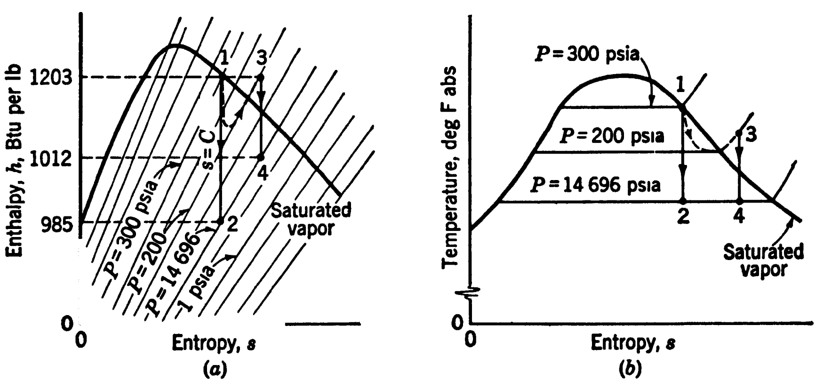


FIG 165 Illustration of decrease in available energy due to throttling.

Although no energy is lost by throttling, availability is decreased as illustrated in Fig. 165. This figure shows saturated steam which may be expanded isentropically from point 1 to atmospheric pressure

at point 2 with maximum energy $(h_1 - h_2)$ available, or 218 Btu for the conditions indicated. If the governor first decreases the pressure to 200 psia by throttling [point 1 to point 3 in Fig. 165(*a*)], the maximum energy available is then reduced to $(h_3 - h_4)$, or 191 Btu for isentropic expansion.

In the real machine both points 2 and 4 would be shifted to the right because of irreversibility in the nozzles and blades with an unavoidable increase in entropy.

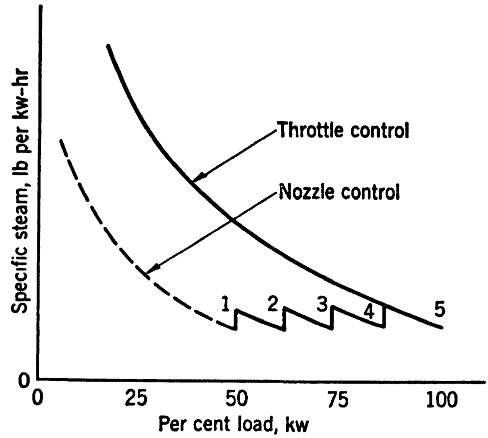


FIG. 166. Comparison of throttle- and nozzle-type governors.

Figure 166 shows a comparison of the specific steam consumption for the two types of control. It can be seen that the nozzle control requires less steam. As additional nozzles are opened with increased load (points 1, 2, 3, and 4) the steam consumption becomes the same as for throttle control.

218. Noncondensing Turbines. Condensers (see Chapter 12) to recover the condensate as feedwater and also to lower the exhaust pressure are used in most plants. There are instances, however, where the use of a condenser is not justified, and in such cases the steam can be expanded only to atmospheric pressure. Such a turbine, known as noncondensing, is found occasionally in older plants or in small plants where fuel is very cheap and good feedwater is plentiful. In reality no cycle is involved because the exhaust steam does not return to the boiler, and such plants are called *noncondensing plants.* The operation as pictured on a *Ts* plane is shown in Fig. 167. The energy in Btu per pound of steam supplied by the boiler is shown by the crosshatched area 6–3–4–5–1–2–7–6 and is equal to the enthalpy change $(h_1 - h_4)$. The

energy that may be extracted as work by an ideal turbine is $(h_1 - h_2)$. At point 2 the steam would be exhausted to the atmosphere, and the remaining energy per pound of steam is represented by the double-crosshatched area 6–3–2–7–6. The energy that may be extracted as

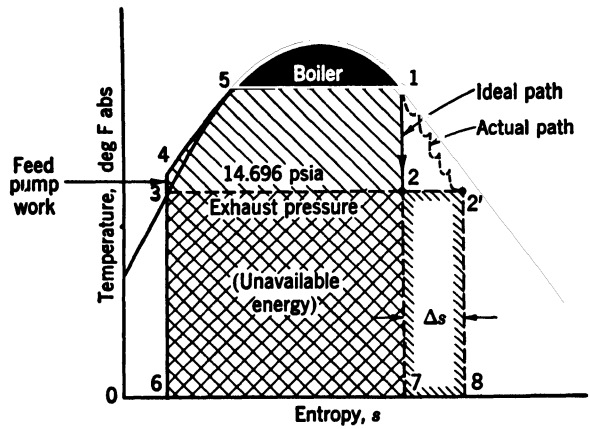


FIG. 167. *Ts* diagram of noncondensing plant.

work is represented by the area 3–4–5–1–2–3 with the exception of a small amount of work done by the boiler feed pump, which is

$$\frac{W}{J} = h_4 - h_3 = \frac{v_f(P_4 - P_3)}{J} \quad \text{Btu per lb} \tag{219}$$

where v_f = specific volume of the boiler feedwater, cu ft per lb.
$(P_4 - P_3)$ = pressure rise across boiler feed pump, psf.

In equation 219 the first equality is applicable for any pressure. The last equality assumes no compressibility of the fluid. The feed pump work is usually negligible for boiler pressures below 400 psia. If feed pump work is neglected, the cycle efficiency is expressed as

$$e_R = \frac{h_1 - h_2}{h_1 - h_{f,2}} \tag{220}$$

as given in Art. 198, where e_R = the Rankine-cycle efficiency and is used as the standard of comparison.

The Rankine-cycle efficiency represents the theoretical maximum that may be obtained from a given turbine-boiler combination. This maximum efficiency is a function of the boiler pressure, the exhaust back pressure, and the quality or degree of superheat. The ratio of the

actual thermal efficiency to the Rankine-cycle efficiency is called the *engine efficiency* (e_e). Engine efficiency indicates how closely the actual efficiency approaches the ideal.

The actual expansion is not isentropic, and the steam passing through the tuıbine will follow a condition curve, point 1 to point 2′ in Fig. 167. As a result of friction effects, there is an increase in the unavailable energy as shown by the area 2–2′–8–7–2 and a corresponding increase in entropy as indicated by Δs.

Steam comsumption for a steam turbine is calculated by the same equation as that for a steam engine, Art. 198. Thus, theoretical steam consumption is

$$m_t = \frac{2545}{h_1 - h_2} \text{ lb per hp-hr} \tag{221}$$

Likewise, Art. 201, actual thermal efficiency is

$$e_t = \frac{2545}{m_s(h_1 - h_{f,2})} \tag{222}$$

where m_s = actual specific steam consumption; and, Art. 202, engine efficiency is

$$e_e = \frac{e_t}{e_R} = \frac{2545}{m_s(h_1 - h_2)} \tag{223}$$

219. Condensing Cycle. If the exhaust steam is discharged into a condenser having a low back pressure, more energy can be extracted per pound of steam as indicated by Fig. 168, which is the same as Fig. 167 but with more complete expansion. The area representing energy converted to work has been increased by that area between the atmospheric pressure line and the 1-in. mercury absolute pressure line. The expressions for efficiencies and steam rate are as stated for the non-condensing unit.

Example. Steam is supplied to a turbine at 200 psia and 700 total temperature. The condenser pressure is 2.0 in. mercury abs. Find (*a*) Rankine-cycle efficiency, neglecting pump work, (*b*) ideal steam rate in pounds per horsepower-hour, (*c*) actual steam rate if the engine efficiency is 54 per cent, and (*d*) thermal efficiency.

Solution. $p_1 = 200$ psia, $h_1 = 1373.6$ Btu, $s_1 = 1.7232$,

$t_1 = 700$ F, $p_2 = 2.0$ in. mercury abs

(*a*) The Rankine-cycle efficiency requires that h_1 and $h_{f,2}$ be found.

$$s_1 = s_2$$

$$s_1 = s_{f,2} + x(s_{fg,2})$$

$$1.7232 = 0.1316 + 1.8481x$$

$$x = 0.862, \quad \text{or} \quad 86.2 \text{ per cent quality}$$

$$h_2 = h_{f,2} + xh_{fg,2} = 69.1 + (0.862)(1036.6) = 963.1$$

$$\frac{h_1 - h_2}{h_1 - h_{f,2}} = \frac{1373.6 - 963.1}{1373.6 - 69\ 1} = 0.315, \quad \text{or} \quad 31.5 \text{ per cent}$$

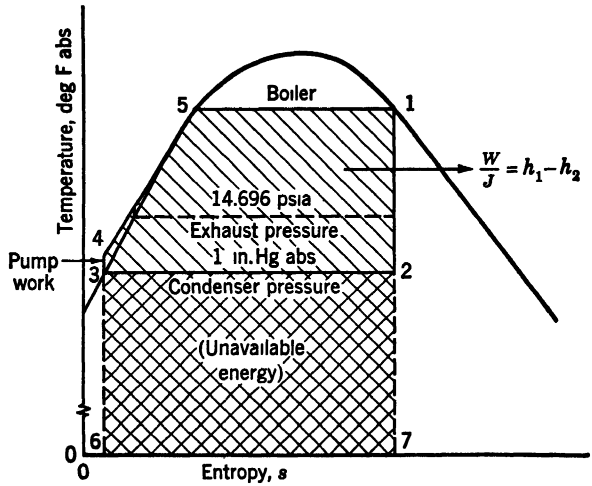


FIG. 168. *Ts* diagram of condensing plant.

(*b*) Ideal steam rate, $m_t = \dfrac{2545}{h_1 - h_2} = \dfrac{2545}{1373.6 - 963.1}$
$= 6\ 20$ lb per hp-hr

(*c*) Actual steam rate, $m_s = \dfrac{6.20}{0\ 54} = 11.48$ lb per hp-hr

(*d*) Actual thermal efficiency, $e_t = \dfrac{2545}{m_s(h_1 - h_{f,2})}$
$= \dfrac{2545}{11.48(1373.6 - 69.1)}$
$= 0.170$, or 17 per cent

220. Regenerative Cycle. Figure 169(*a*) shows a diagrammatic layout of a condensing steam power plant in which a surface condenser is used to condense all steam that is not extracted for feedwater heating. The turbine is double extracting, and the boiler is equipped with a superheater. The cycle diagram would appear as shown in Fig. 169(*b*). This arrangement constitutes a *regenerative cycle.* It is widely used to

improve efficiency and to provide steam for evaporating make-up feedwater.

The ideal regenerative efficiency can be expressed as the isentropic work divided by the energy input. For two-stage extraction the regenerative cycle efficiency, neglecting pump work, is

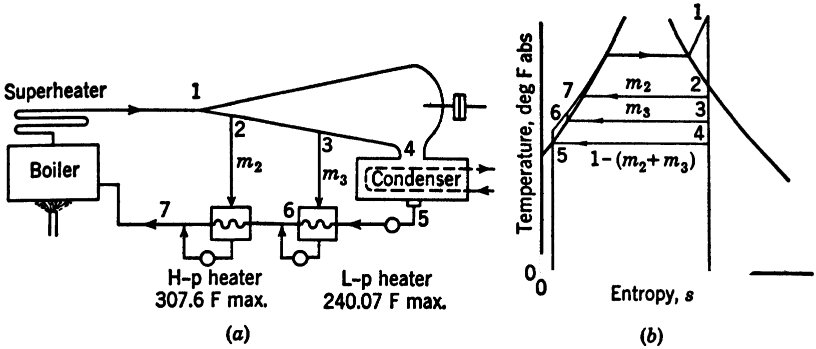


FIG. 169. Double-extraction regenerative cycle.

$$e_r = \frac{m_2(h_1 - h_2) + m_3(h_1 - h_3) + [1 - (m_2 + m_3)](h_1 - h_4)}{h_1 - h_{f,7}} \tag{224}$$

where m_2 = pounds of high-pressure steam extracted per pound of steam flow.

m_3 = pounds of low-pressure steam extracted per pound of steam flow.

$1 - (m_2 + m_3)$ = pounds of steam entering condenser per pound of steam flow.

The quantities of steam m_2 and m_3 to be extracted in order to heat the feedwater can be determined by making energy balances. The product of the weight of water to be heated and its enthalpy change must equal the product of the weight of the steam to be extracted and its enthalpy change. Thus, from Fig. 169,

$$m_3(h_3 - h_{f,6}) = [1 - (m_2 + m_3)](h_{f,6} - h_{f,4}) \tag{225}$$

$$m_2(h_2 - h_{f,7}) = (1 - m_2)(h_{f,7} - h_{f,6}) \tag{226}$$

All enthalpies may be determined; therefore m_2 and m_3 may be found. The maximum temperature to which the water can be heated is dictated by that of the bled steam. The condensate from the bled steam is added to the feedwater.

Example. A double-extraction turbine, Fig. 169, operates on an ideal regenerative cycle, initial pressure 200 psia, total temperature 450 F, high-pressure extraction at 75 psia, low-pressure extraction at 25 psia, condenser pressure of 1 in. mercury abs. Neglect pump work. Find, per pound of steam flow, (*a*) the low-pressure extraction, (*b*) the high-pressure extraction, and (*c*) the regenerative cycle efficiency.

Note: The solution may be expedited by using a Mollier chart to determine enthalpies during isentropic expansion.

Solution. See Fig. 169 for symbols and state point references. From the Mollier chart.

$$h_1 = 1241 \text{ Btu per lb} \quad h_3 = 1075 \text{ Btu per lb}$$

$$h_2 = 1153 \text{ Btu per lb} \quad h_4 = 892 \text{ Btu per lb}$$

From steam tables,

$$h_{f,2} = h_{f,7} = 277.4; \quad h_{f,3} = h_{f,6} = 208.4; \quad h_{f,4} = h_{f,5} = 47.05 \text{ Btu per lb}$$

(*a*)

$$m_2(h_2 - h_{f,7}) = (1 - m_2)(h_{f,7} - h_{f,6})$$

$$m_2(1153 - 277.4) = (1 - m_2)(277.4 - 208.4)$$

$$m_2 = 0.073 \text{ lb per lb of flow}$$

(*b*)

$$m_3(h_3 - h_{f,6}) = [1 - (m_2 + m_3)](h_{f,6} - h_{f,5})$$

$$m_3(1075 - 208.4) = [1 - (0.073 + m_3)](208.4 - 47.05)$$

$$m_3 = 0.1455 \text{ lb per lb of flow}$$

(*c*) Substituting in equation 224,

$$e_r = \frac{0.073(1241 - 1153) + 0.1455(1241 - 1075) + (1 - 0.073 - 0.1455)(1241 - 892)}{(1241 - 277.4)}$$

$e_r = 0.3142$, or 31.42 per cent

Example. Find the Rankine-cycle efficiency of a turbine operating at the same pressures and temperatures as given in the above example but with no extraction.

Solution.

$$e_R = \frac{h_1 - h_4}{h_1 - h_{f,5}} = \frac{1241 - 892}{1241 - 47.05} = 0.2921, \quad \text{or} \quad 29.21 \text{ per cent}$$

221. Reheat Cycle. A system in which steam is partially expanded through a turbine and then reheated before further expansion in the turbine is called a *reheat cycle*. The cycle permits the expansion of steam from high pressures (1000 psig and up) to low exhaust pressures without excessive condensation. Usually the steam is expanded through one cylinder of a compound turbine and then reheated before expanding through the remaining cylinders, as in Fig. 170, which illustrates both reheat and regeneration.

The effect of reheating is illustrated by the temperature-entropy di-

agram, Fig. 171. In this cycle steam is assumed to expand isentropically from point 1 to point 2 at which it is returned to the furnace and reheated to point 3, then expanded to point 4 in the turbine. The

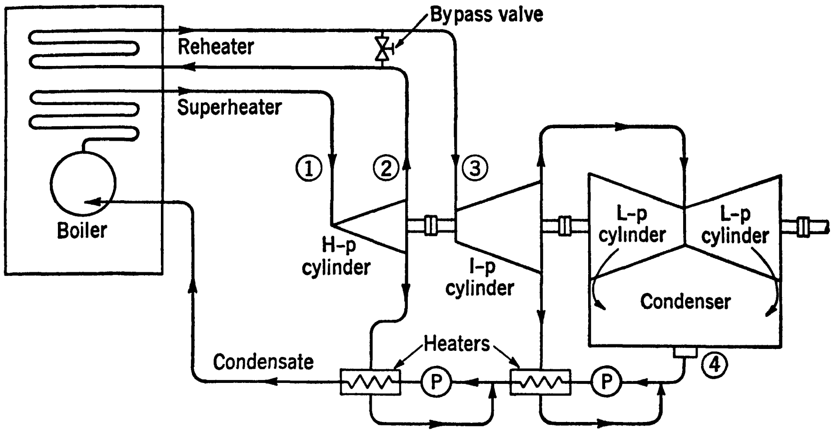


FIG. 170. Flow diagram for reheat and regeneration.

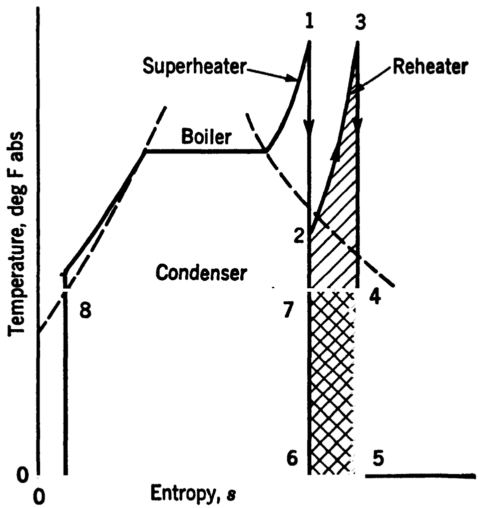


FIG. 171. *Ts* diagram for a reheat cycle.

energy supplied during reheating is shown graphically as area 2–3–5–6–2, the available part of this energy as area 2–3–4–7–2, and the unavailable part as area 4–5–6–7–4. The efficiency of the ideal reheat cycle is the ratio of the available energy to that supplied. If this ratio exceeds the

cycle efficiency obtained without reheating, then a net gain is obtained. The gain may improve the over-all efficiency from 3 to 6 per cent. The rapid increase in specific volume of steam with decreasing pressure requires that the reheating be done at pressures usually exceeding 200 psig.

Figure 172 shows a three-cylinder tandem-compound 150,000-kw turbine designed for reheat. Steam conditions are 1800 psig and 100 F with reheating to 1000 F after expansion through the high-pressure

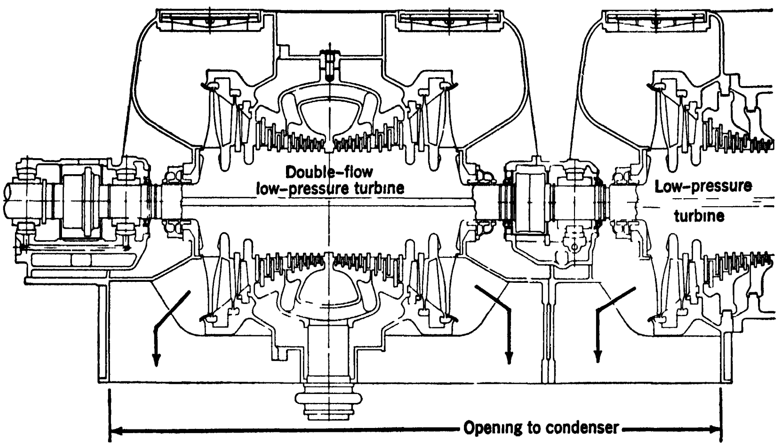
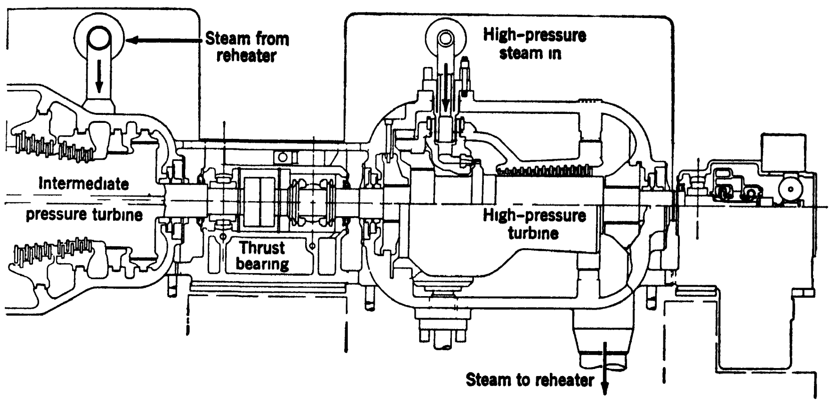


FIG. 172. Allis-Chalmers proposed

cylinder. Steam flow in the high-pressure cylinder is in the opposite direction to that in the intermediate-pressure cylinder. Thus high-pressure steam and reheated steam enter the turbine in such a way that expansion of the turbine shaft tends to equalize in both directions since the thrust bearing is located between these two cylinders. Blades, 23 in. long, are used in the last row of all three low-pressure flow paths.

222. Mercury-Steam Binary Vapor Cycle. Mercury has many advantages as a thermodynamic fluid for use in power generation. Its vapor pressure is low at high temperatures, being 140 psig at 975 F and 300 psig at 1100 F. Its critical temperature is 2240 F, and its freezing point −38 F. Since mercury is an element, it is stable at temperatures well above the limit imposed by materials now available for boiler tubes, piping, and turbine nozzles. The combination of two media operating in separate circuits is called a *binary vapor cycle*, and such cycles approach the ideal Carnot efficiency for the temperature limits now possible with available blade materials. Experience has shown that steam is the best medium available for use as the lower-temperature fluid in a binary vapor cycle.

New power plants have been designed specifically for use of the binary vapor system; however, a greater application has been as topping units. An installation at South Meadow, Hartford, Conn.,[1] is illustrated in Fig. 173, which shows a 15,000-kw mercury unit serving as a topping turbine for the main steam plant. The operating conditions for the mercury system, which contains 180,000 lb of mercury, are 113 psig and 945 F.



150,000-kw reheat steam turbine.

The mercury boiler is designed to deliver 1,640,000 lb per hr of mercury vapor; it is a single-drum type with 668 tubes (1⅝-in. O.D.) lining the firebox, which is 29 ft by 20½ ft by 44 ft in height. The furnace is fired by six oil burners. The outlet for combustion gases is equipped with a conventional steam superheater, economizer, and regenerative air heater followed by an induced-draft fan.

The mercury turbine is a five-pressure-stage impulse machine which operates at 720 rpm with a single horizontal two-bearing shaft, the turbine wheels overhanging with the highest-pressure wheel at the outer end. The mercury is delivered to the high-pressure wheel through a scroll in which the first-stage nozzles are located with conventional diaphragms placed between successive wheels. The turbine exhausts to the mercury condenser, from which the mercury returns to the mercury boiler. Water coming from the economizer cools the mercury condenser and is converted to saturated steam at 410 psig and 450 F. This steam is returned to the superheater and heated to 700 F, after which

[1] "South Meadow—Mercury Power Unit," O. L. Wood, *Power Generation*, March, 1950, pp. 62–67.

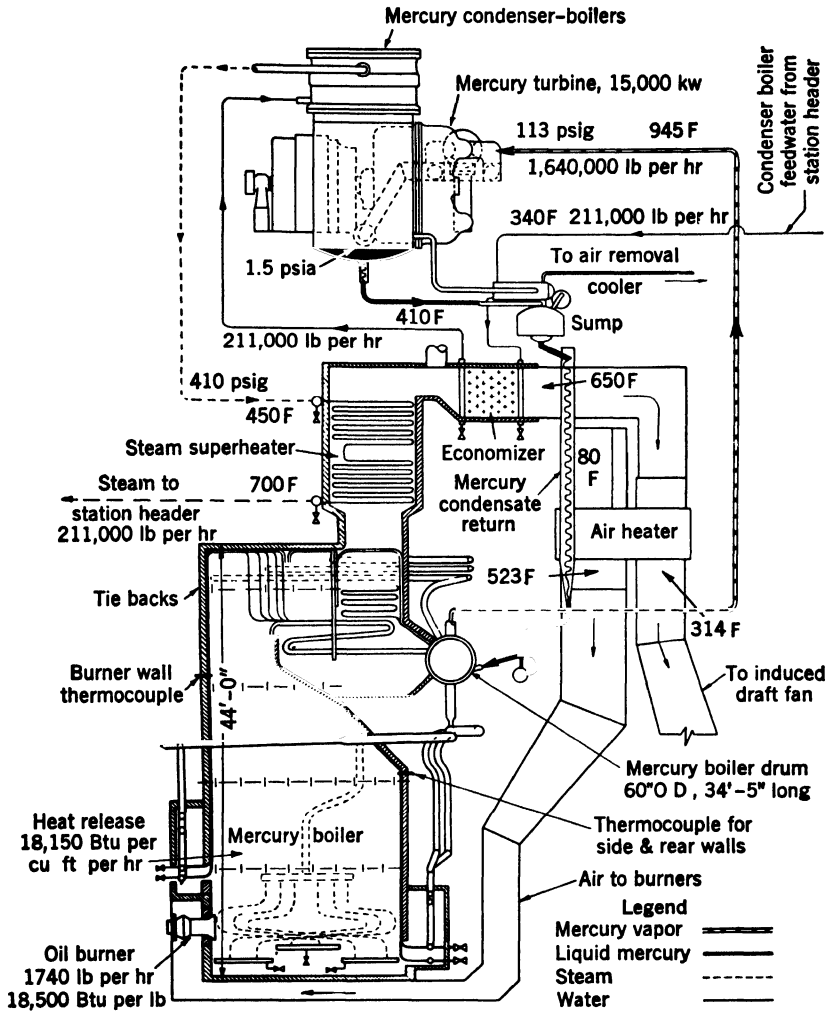


FIG. 173. General Electric mercury boiler and turbine.

it is used in the conventional steam turbines. The difference in temperature between the condensing mercury and the boiling water averages 39 F, and the excellent wetting properties of the mercury provide good heat transfer at this temperature difference.

Treatment of the boiler mercury with metallic magnesium and titanium hydride has been found necessary. The magnesium constitutes a deoxidizer for cleaning the mercury since it has a great affinity for oxygen, the resulting oxide being trapped as dust near the condenser

pump. The titanium forms an alloy steel layer on the tube surfaces, thus preventing the solution of iron by the mercury. The two additives provide a mercury silvering or wetting of the tube surfaces, and the rate of heat transfer between tube and mercury is greatly increased.

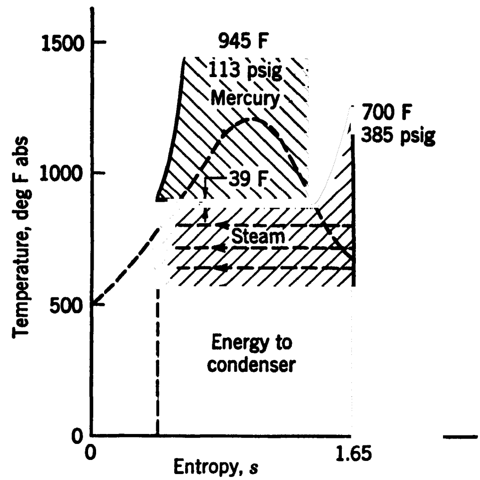


FIG. 174. *Ts* diagram of binary-vapor cycle.

Figure 174 shows a *Ts* diagram of the system. Assuming isentropic expansion, the mercury system has an efficiency of approximately 31.5 per cent, and the steam system 27 per cent, making a combined efficiency of 58.5 per cent. The Carnot-cycle efficiency for the same temperature limits is approximately 63 per cent.

223. Trends in Turbine Construction. Since 1948 the capacity of the largest 3600-rpm single-shaft turbine built has increased yearly on the average of nearly one fifth.[2]

Two turbines of 250,000-kw capacity each are being built for TVA installation in 1955. Engineers see no reason why this figure cannot be exceeded. Inlet steam temperatures of 1050 to 1100 F are possible today. Pressures of 1450 and 1800 psig are common. A few turbines use steam at approximately 2400 psig, and one at 4500 psig. The need for higher pressures has been eased by the rapid adoption of the reheat cycle.

Figure 175 shows the assembling of the low-pressure rotor of a 50,000-kw three-cylinder steam turbine, typical of the design trends for turbines of this size.

An important development is the 2-in. increase in length of last-row

[2] *Westinghouse Engineer*, Jan., 1953, p. 2.

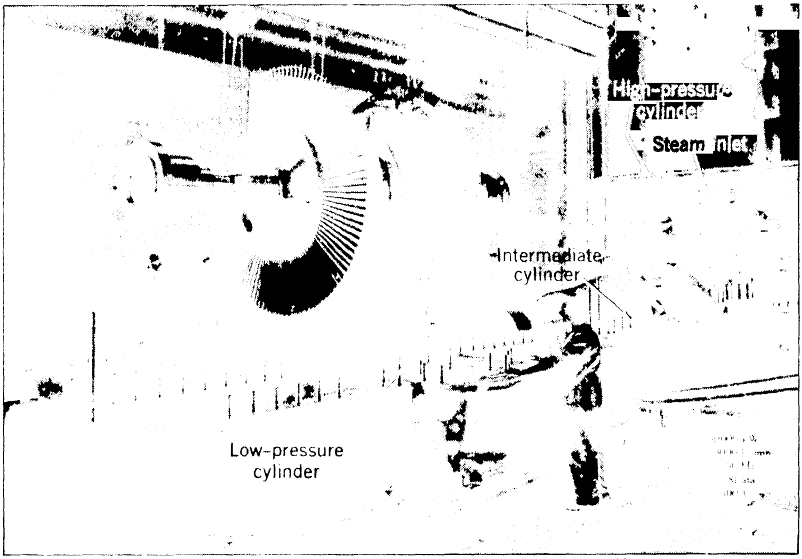


FIG. 175. Lowering the low-pressure rotor of a Brown Boveri 50,000-kw three-cylinder steam turbine into its casing for testing.

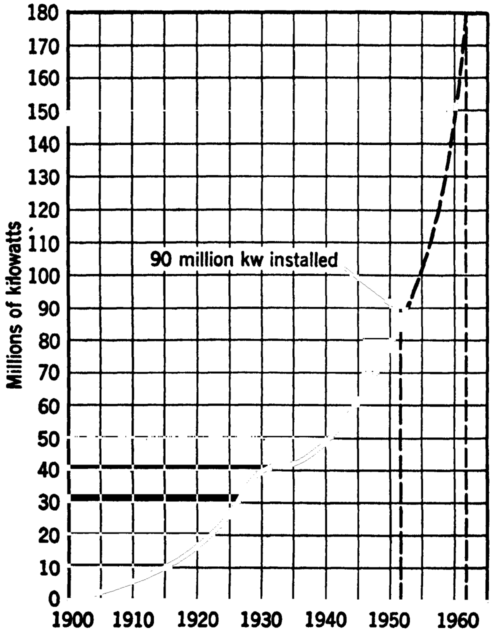


FIG. 176. Growth of generating capacity of steam turbines in the United States.

blades for 3600-rpm turbines. These blades, now 25 in. long, are mounted on a wheel of 20 per cent greater diameter and provide for 25 per cent more steam flow exhaust area. In effect, the maximum output of high-speed turbines has been increased 25 per cent with no increase in the energy remaining in the steam as it enters the condenser. The tip speed of these blades, mounted on a 50-in.-diameter spindle, is 1073 mph. High-strength materials and methods of controlling blade vibration have made possible this development.[3]

Figure 176 shows the increase in generating capacity since 1900. What comes next is uncertain. Perhaps hollow blades, to keep centrifugal forces within reason, may appear.

PROBLEMS

1. An ideal nozzle receives steam at an initial pressure of 300 psia and a total temperature of 600 F. Discharge pressure is 50 psia. Assuming isentropic expansion, find the exit velocity, fps.

2. An ideal steam nozzle expands 5400 lb of steam per hr from an initial pressure of 200 psia and a total temperature of 500 F to a back pressure of 40 psia. Find in inches (*a*) throat diameter; (*b*) exit diameter; and (*c*) nozzle length.

3. Calculate the weight of steam, in pounds per hour, that can be delivered by an ideal nozzle which has a throat area of 0.5 sq in., when dry saturated steam is supplied at 300 psia. 7490 lb per hr, *Ans.*

4. A steam turbine delivers 1 kw-hr for every 13 lb of steam supplied at 200 psia and 440 F. The exhaust back pressure is 1 in. mercury absolute. Find the thermal efficiency and the Rankine-cycle efficiency.

5. A turbine is supplied with dry saturated steam at a pressure of 400 psia when the condenser pressure is 1 in. of mercury absolute. There are three pressure stages so designed that 40 per cent of the work is done in the first stage, 35 per cent in the second, and 25 per cent in the third stage. What is the approximate pressure drop in each stage, assuming isentropic expansion?

6. A steam engine receives steam at 200 psia and 400 F and exhausts at 20 psia. If a low-pressure turbine exhausting at 1 in. of mercury absolute is added to utilize this exhaust steam, what would be the percentage of increase in energy available, assuming isentropic expansion? 115 per cent, *Ans.*

7. Dry saturated steam is expanded in an ideal turbine nozzle at the rate of 3600 lb per hr from a pressure of 200 psia to 14.696 psia. (*a*) If the turbine blades transform 80 per cent of the kinetic energy into work, determine the horsepower developed. (*b*) For maximum work, determine the rpm of a single wheel if the mean diameter is 36 in. Neglect losses, and assume zero nozzle angle with respect to turbine blade.

8. A single-stage impulse turbine receives dry saturated steam at a pressure of 180 psia and exhausts at 4 psia. The peripheral speed of the blades is 1400 fps, and the nozzle angle is 20 deg. (*a*) Assuming isentropic flow, draw to scale a velocity diagram showing the blade angle, in degrees, and the absolute exit velocity of the steam, fps. (*b*) Find the theoretical work done in foot-pounds if the flow rate is 1 lb per sec. (*c*) Calculate the stage efficiency of the turbine.

[3] *Westinghouse Engineer*, May, 1952, pp. 96–97.

9. An ideal turbine receives 5400 lb of steam per hr at 200 psia and 400 F and exhausts at 2 in. mercury absolute. The wheel has a mean diameter of 36 in. and operates at 3600 rpm. Determine (*a*) the number of velocity stages required, (*b*) pressure stages required, and (*c*) the probable number of reaction stages required.

10. Assume that the total steam consumption of a throttle-governed turbine varies as a straight line with respect to its load. A 21,000-kw turbine generator is supplied with steam at 400 psia and 600 F, and its exhaust pressure is 1 psia. At its rated load the engine efficiency is 60 per cent. At no load the turbine uses 25,000 lb of steam per hr. What will be the unit steam consumption, pounds per kilowatt-hour, at two-thirds load?

11. A steam turbine generator is supplied with steam at 315 psia and 680 F abs and operates with an exhaust pressure of 1.5 in. mercury absolute. Its unit steam rate at 24,000-kw load is 9.8 lb per kw-hr. Generator efficiency is 92 per cent. Find (*a*) the thermal efficiency for the turbine, (*b*) the Rankine-cycle efficiency, and (*c*) the engine efficiency for the turbine as a unit.

12. A turbine generator uses 18 lb of steam per kw-hr delivered at the switchboard. Steam pressure, 300 psia; total temperature, 550 F; vacuum in jet condenser, 27 in. mercury; barometric pressure, 29 in. mercury absolute; mechanical and electrical efficiency of the combined unit, 90 per cent; temperature of inlet water to jet condenser, 57 F; temperature of discharged water from condenser, 85 F. Find: (*a*) thermal efficiency based on kilowatt-hours delivered at switchboard, (*b*) quality of the steam at exhaust (neglecting radiation losses), and (*c*) theoretical weight of condenser water required per pound of steam.

(*a*) 15.57 per cent, (*b*) 97.20 per cent, (*c*) 36.60 lb per lb of steam, *Ans.*

13. The following data apply to a turbine-driven electric generator: total steam consumption, 25,000 lb per hr; steam pressure at throttle, 300 psia; steam temperature at throttle, 600 F; generator efficiency, 90 per cent; condenser pressure, 1.473 psia; output at switchboard, 1800 kw. Find: (*a*) thermal efficiency (referred to turbine coupling); (*b*) Rankine-cycle efficiency; (*c*) engine efficiency; and (*d*) steam rate referred to turbine coupling.

14. One pound per second of dry saturated steam at 200 psia expands isentropically from zero initial velocity to a back pressure of 5 psia. Find: (*a*) quality of steam at exit; (*b*) exit velocity, fps; and (*c*) theoretical horsepower available.

15. A steam turbine operating at a load of 30,000 kw requires 15 lb of steam per kw-hr delivered at the switchboard. The steam is supplied at 200 psia and 500 F. The temperature of the feedwater entering the boiler is 184 F, and the boiler efficiency is 78 per cent. (*a*) How many pounds of coal are required per kilowatt-hour if the coal has a heating value of 13,500 Btu per lb? (*b*) How many boilers would be necessary to carry the load if each of the boilers supplying the steam has 20,000 sq ft of heating surface and operates at 150 per cent of rating? (*c*) What would be the over-all efficiency of the plant?

16. Determine the number of theoretical reaction stages for a steam turbine to operate at 3600 rpm when supplied with 6000 lb of steam per hr at a pressure of 260 psia and a temperature of 500 F. The steam is expanded to an exhaust pressure of 0.982 psia. The mean diameter of the rotor is 48 in.

17. A steam turbine operates at a speed of 3600 rpm. The mean diameter of the rotor blades is 48 in. The steam-inlet conditions are 450 psia and 550 F, and the condenser pressure is 0.74 psia. Assuming isentropic expansion, determine the number of ideal stages required for each of the following cases: (*a*) all velocity stages; (*b*) all pressure stages; (*c*) first stage, two-row velocity; remaining stages, Rateau; and (*d*) all reaction stages.

18. A turbine generator delivers 2000 kw and is supplied with 12 lb of dry steam per kw-hr; steam pressure, 200 psia; and exhaust, 1.23 psia. The boiler supplying steam to the above turbine receives feedwater at 182 F. The evaporation is 9 lb of water per lb of coal as received. Coal as received contains 12,000 Btu per lb. Find (*a*) the over-all thermal efficiency of the plant and (*b*) the thermal efficiency of the boiler.

19. A compound turbine receives steam at 1200 psia and 900 F. After expansion to a temperature of 500 F in the high-pressure cylinder the steam is reheated to 900 F and then expanded in a low-pressure cylinder to an exhaust pressure of 0.491 psia. Assume isentropic expansion. (*a*) Find the exhaust pressure of the high-pressure cylinder. (*b*) What is the quality of the steam at the low-pressure cylinder exhaust? (*c*) What percentage of the total power of the turbine is developed in the high- and in the low-pressure cylinders? (*d*) Find the reheat cycle efficiency.

20. A turbine is supplied with steam at 500 psia and 600 F; exhaust is at 0.74 psia. (*a*) Find the ideal regenerative cycle efficiency if steam is extracted at a pressure of 40 psia. (*b*) Find the Rankine-cycle efficiency, assuming no extraction. Neglect pump work.

21. A turbine is supplied with steam at 500 psia and 700 F. Steam is extracted from the turbine at 150 psia and at 40 psia to heat feedwater. The exhaust pressure of the turbine is 1 psia. For the ideal regenerative cycle calculate (*a*) the theoretical weight of steam to be extracted at each point per pound of steam supplied to the turbine and (*b*) the regenerative cycle efficiency. (*c*) Determine the Rankine-cycle efficiency when the turbine is operated condensing and no steam is extracted. Neglect pump work.

CHAPTER 12

Steam-Condensing Equipment

224. Applications of Condensers. Steam condensers are devices in which the exhaust steam from engines and turbines is condensed and air and other noncondensable gases are removed in a continuous process. Two advantages may be gained through the use of condensers in steam-turbine and steam-engine operation: (1) reduction of the exhaust pressure, with consequent increase in available energy, and (2) recovery of the condensate for re-use as boiler feedwater. For most steam plants recovery of the condensate is very important, and for most marine applications it is a necessity. Boiler feedwater taken from a river, lake, or ocean must be evaporated or treated before being used in a boiler. With trends toward higher pressures and temperatures in boilers the need of pure feedwater has increased, with the result that most condenser installations are of the surface type which permits recovery of the condensate.

The Mollier diagram, Fig. 39, shows that the energy theoretically available by expanding steam isentropically from atmospheric pressure and saturation (14.696 psia and 212 F) to 0.2 psia is 250 Btu per lb. This energy is essentially equal to the energy available from steam at 175 psia and 750 F if expanded isentropically to atmospheric conditions. This fact makes the use of low-pressure turbines practicable. Many early installations that were originally designed as noncondensing plants have been rejuvenated by installing condensers and low-pressure turbines. The thermal efficiency and capacity of the plant may be greatly increased and the condensate salvaged. Practical aspects, however, such as the amount, temperature, elevation, and cleanliness of a cooling-water supply largely determine whether a condenser installation is feasible.

The condensation of steam in a closed vessel produces a partial vacuum by reason of the great reduction in the volume of the low-pressure vapor. A pound of dry steam at 14.7 psia has a volume of 26.80 cu ft. Theoretically, if this quantity were contained in a steam-tight vessel of 26.80 cu ft volume at 14.7 psia and if condensation into

water occurred at 142 F, the liquid would occupy only 0.0163 cu ft. The volume would be 1/1644 of the space inside the vessel, and the pressure would fall to 3 psia. The theoretical power necessary to operate such a condenser would be that required to pump the pound of liquid condensate from 3 psia to 14.7 psia plus that necessary to circulate the cooling water.

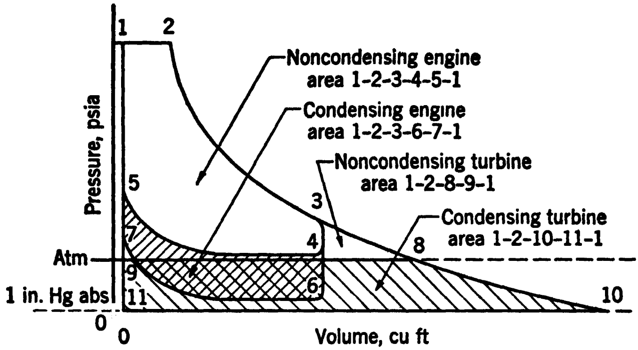


FIG. 177. Effect of condenser on ideal work diagram.

Figure 177 illustrates the increase in work possible by the use of a condenser. Steam turbines are capable of expanding the steam to the lowest exhaust pressure obtainable because they are steady-flow machines and may have large exhaust outlets (with no valves) through which the spent steam must be discharged. Steam engines, however, are intermittent-flow machines and must force the expanded steam out the relatively small exhaust valves. The law of diminishing returns fixes the point at which the friction losses in the necessarily large cylinder plus the work of discharging the exhaust exceed the gains derived from low exhaust pressure. The lowest practical exhaust pressure for most steam engines is therefore 6 to 8 in. mercury absolute. Steam turbines may expand steam to 1 in. mercury absolute pressure or less.

In practice, a larger power requirement may be needed to remove the air and noncondensable gases which enter the condenser and which cannot be removed by condensation. Leakage of air into the condenser may occur at the turbine shaft, gaskets, atmospheric vent valve, and other points. Some air and gases are carried in solution by water from natural sources, and under the low pressure of the condenser these gases tend to be released.

225. Types of Condensers. For power plant use there are two types of condensers: (1) *surface* and (2) *jet*. Surface condensers provide both a low exhaust pressure and recovery of the condensate, whereas jet

condensers provide only the low exhaust pressure and the condensate escapes with the cooling water. In large steam-turbine installations, jet condensers are impractical because, apart from the loss of the condensate, the power consumption of jet-condenser pumps and the first cost of the requisite air pumps outweigh the advantage of the high vacua attained. However, with moderate-sized turbine units and for

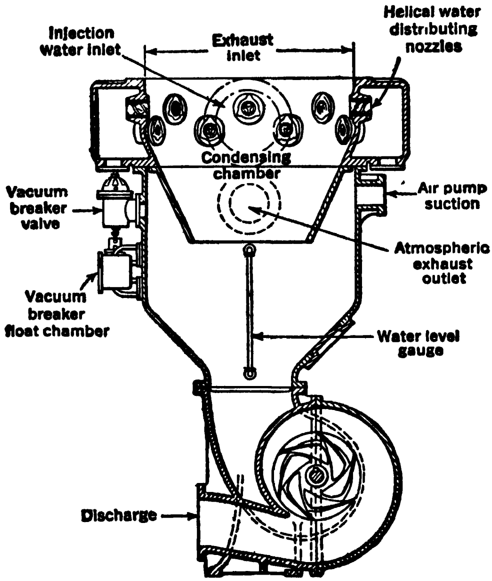


FIG. 178. Low-level jet (direct-contact) condenser.

reciprocating steam engines, the jet type has a field of useful application, especially where an abundant supply of good feedwater is available.

A surface condenser generally consists of a cast-iron or mild-steel shell, fitted with a tube plate at each end. A great number of tubes extend between the tube plates, thus forming the cooling surface. The exhaust steam is admitted to the condenser through an opening in the top of the shell, and cooling water is passed through the tubes. When the condenser is used in conjunction with a pumping engine, the reverse arrangement is generally adopted; i.e., the water passes over the outside of the tubes, and the steam is allowed to pass through the tubes. This arrangement is called a *water-works condenser*. Another form of surface condenser, known as an *evaporative condenser*, has the shell omitted. The steam is fed through the condenser tubes over which water is sprayed. The cooling is effected mainly by evaporation of the water in the atmosphere.

Jet condensers may be classified as *low-level jet* and *barometric condensers*. The two types are similar in the manner in which the exhaust steam and cooling water are brought together but differ in the method of removing the water and the condensate Jet condensers in which the cooling water, condensate, and noncondensable gases are withdrawn by

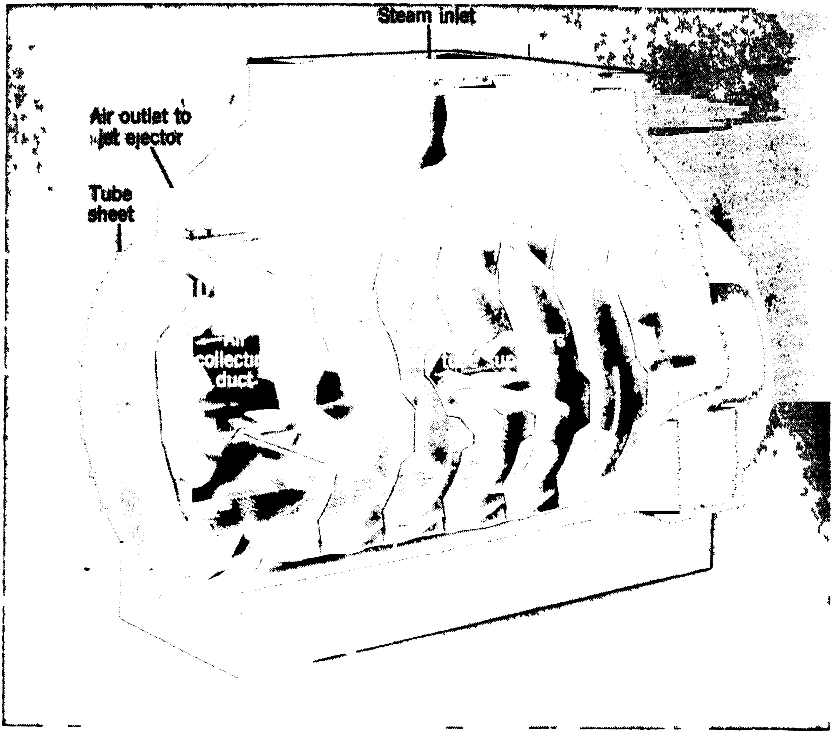


FIG 179. Westinghouse two pass radial flow surface condenser.

a single pump are called *low-level low-vacuum jet condensers* because of the limited air capacity of the pump In the condenser illustrated in Fig. 178, the noncondensable gases are removed by a separate pump or ejector, higher vacua are obtainable, and the condenser is known as a *low-level high-vacuum jet condenser* Under certain conditions the air and water can be removed by the kinetic action of the stream, in which case the jet condenser is designated as an *ejector* or *siphon condenser*.

226. Surface Condensers. Condensate can be recovered from the surface condenser because the steam and circulating water do not mix. The steam to be condensed is normally made to flow over the outside of the tubes, Fig. 179, while the cooling or circulating water is made to

flow inside the tubes. This flow pattern is used mainly because the clean steam does not contaminate the outside of the tubes which would be difficult to clean. The circulating water is frequently dirty and leaves deposits on the inside of the tubes. The usual method of cleaning the inside of the tubes consists of removing the end cover plates from

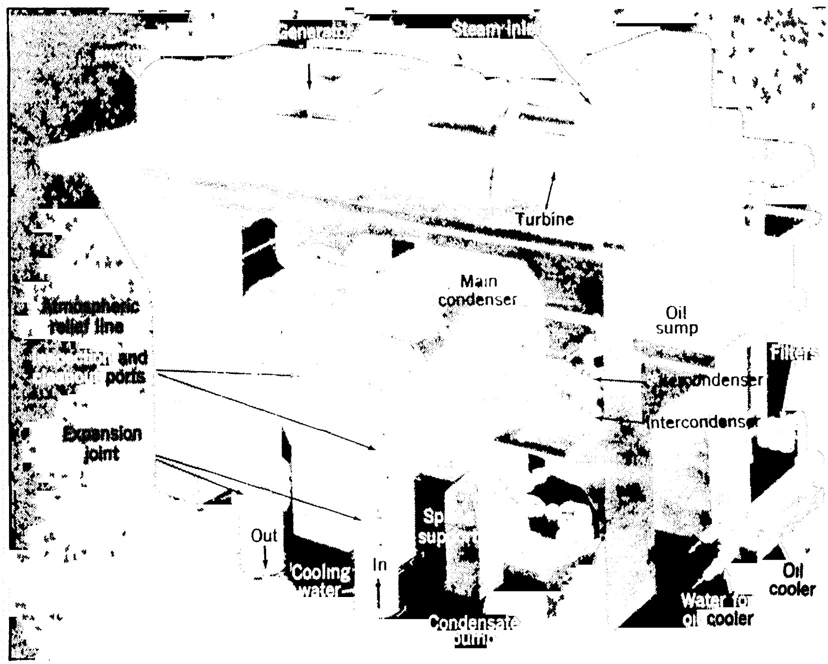


FIG. 180 Elliott condenser and turbine installation.

the condenser and passing motor-driven wire brushes through the tubes. This is no small task since a condenser may have one to eleven thousand tubes.

Surface condensers may be single-pass, in which the water flows in one direction only through all the tubes, or two-pass, in which the water flows in one direction through part of the tubes and returns through the remainder. Most large condensers are equipped with a centrifugal pump for removing the liquid condensate and a jet-type air ejector, Art. 229, is used for removing the air and gases. Figure 180 shows a typical modern turbine and condenser installation. The two-pass condenser is suspended directly from the bottom of the turbine, and no expansion joint is needed. Spring supports aid in carrying the condenser load and allow movement due to expansion and contraction.

The condenser water lines are usually equipped with rubber expansion joints since they are subject only to the low pressure of the circulating water. The condensate pump removes the water as fast as it falls to the hot well. The condensate serves as a coolant for the inter- and after-condensers as it is pumped to the storage tank or low-pressure heater.

The air and noncondensable gases are removed from the main condenser by steam-operated ejectors. As shown in Fig. 180, two ejectors operate in parallel between the main condenser and the intercondenser. These ejectors move the air from the main condenser to the intercondenser where the pressure is approximately 7 psia. Two other ejectors operate in parallel to move the air from the inter- to the aftercondenser which is at atmospheric pressure. The air and noncondensable gases are therefore compressed in two stages with a pressure rise of about 7 psi each as they are discharged to the atmosphere. The high-pressure steam used to operate the ejectors is condensed in the inter- and after-condensers and is usually drained through traps and returned to the feedwater system.

A surface condenser and its auxiliaries should accomplish the following results:

1. Steam should enter the condenser with the least possible resistance, and the pressure drop through the condenser should be reduced to a minimum.

2. Air, a poor conductor of heat, should be rapidly removed from the heat-transmitting surfaces.

3. The air should be collected at suitable places, practically free from entrained water vapor and cooled to a low temperature.

4. The air removal should be done with a minimum expenditure of energy.

5. Condensate should also be rapidly cleared from the heat-transmitting surfaces, freed from air, collected for removal, and returned to the boiler at the maximum practicable temperature.

6. Circulating water should pass through the condenser with little friction, with minimum deposit of sediment, and with maximum heat absorption.

Condenser tubes are usually made of red brass (85 copper, 15 zinc), or Muntz metal (60 copper, 40 zinc) for pure water, and Admiralty brass (70 copper, 29 zinc, 1 tin) for salt water and impure river waters. The outside diameter of the tubes varies from 5/8 to 1 in., using No. 18 B.W.G., which has a wall thickness of 0.049 in.

Figure 181 shows a cutaway view of a modern two-pass double-flow condenser designed to require a minimum of head room. Therefore the

turbine foundation may be low and less costly. Figure 182 shows a sectional view of this unit which has 70,000 sq ft of tube surface and can

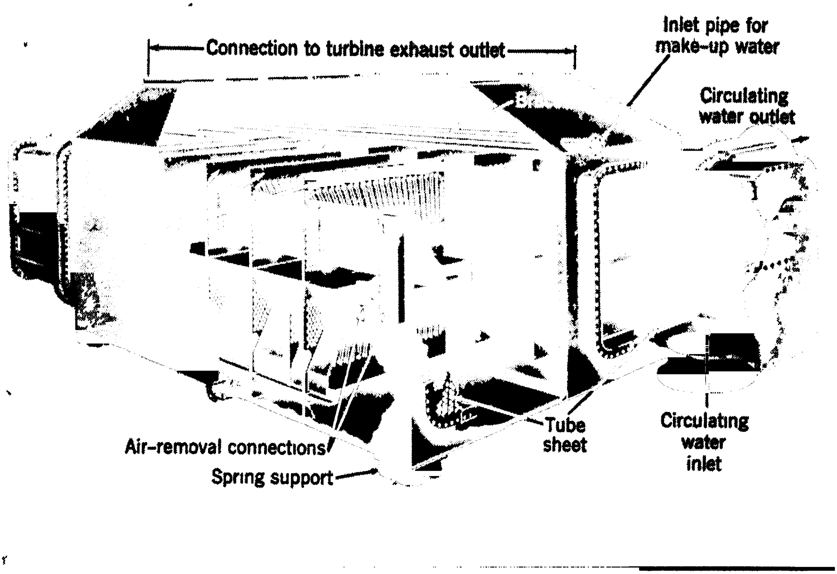


FIG. 181. Cutaway view of Foster Wheeler double-flow condenser.

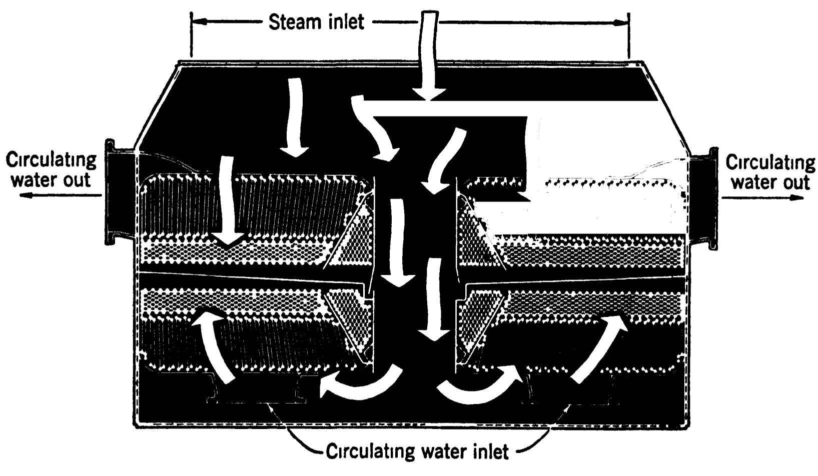


FIG. 182. Transverse sectional elevation of double-flow condenser.

serve a 100,000-kw turbine. The arrows show steam flowing directly downward into the tube bank; at the same time it flows downward

through the center steam lane, then up into the tube bank. This flow pattern gives rise to the term "double flow" and provides inherent deaerating and reheating action. All steam condensing on the lower tubes drips back in counterflow to the incoming steam. The steam condensing on the upper tubes drips through and falls upon the inclined baffle separating the upper and lower tube banks. This condensate then

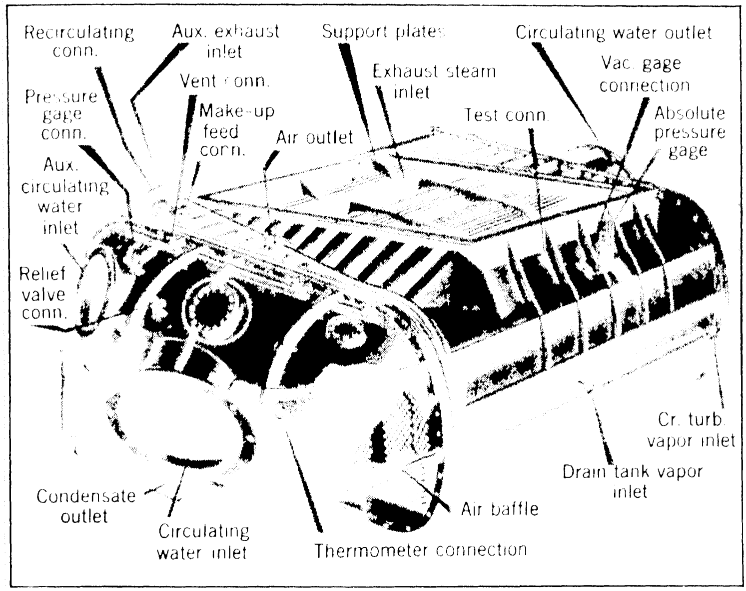


FIG. 183. Foster Wheeler marine surface condenser.

flows through a water seal and downward over the edge of the vertical baffle on which it forms a thin film. As this water film leaves the lower edge of the baffle it falls through the live steam, thus providing deaeration as well as preventing subcooling. The large entrance area and relatively short flow paths into the tube banks aid in reducing friction and pressure loss. The triangular air-cooler section shown in the center of Fig. 182 is used to reduce the volume of air and noncondensable gases before removing them by an air pump.

Figure 183 illustrates a surface condenser designed for marine use. Cooling water, brought in by scoops placed on the outside of the ship's hull, flows through the tubes and is discharged. No circulating-water pump is necessary when the vessel is in motion but an auxiliary pump is provided. Exhaust steam from the turbines enters the top of the condenser, flows down the steam channels, and is condensed on the outside

of the tubes. The condensate is removed at the bottom by means of a pump.

227. Low-Level Jet Condensers. The low-level multijet condenser is illustrated in Fig. 184. The condenser consists of a closed cylindrical

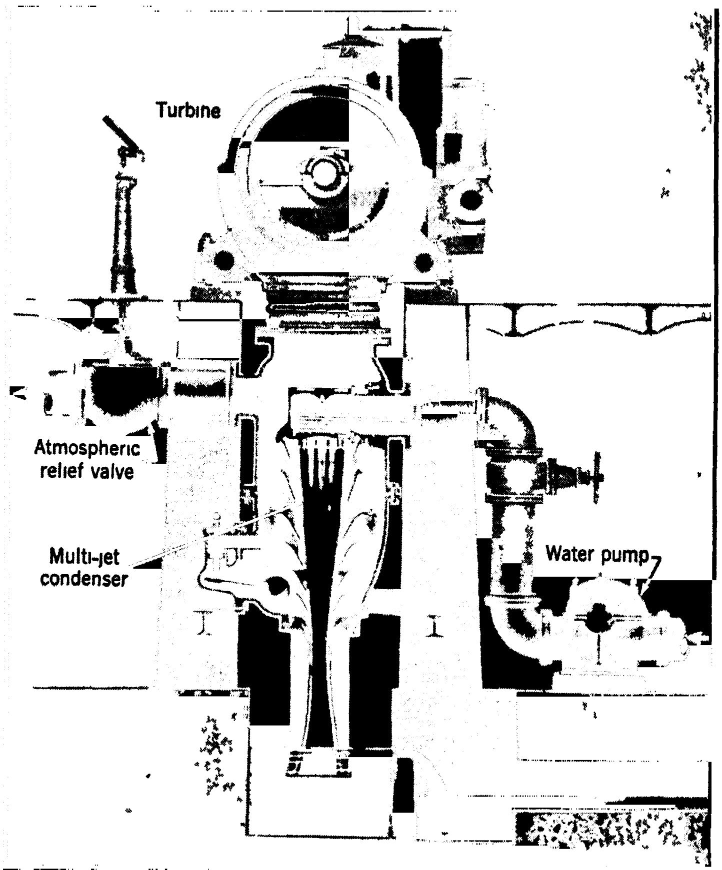


FIG. 184 Schutte and Koerting ejector-condenser installation

chamber in the upper end of which is a water-nozzle case. The water-nozzle case is fitted with a Venturi-shaped combining tube. The combining tube is connected with a tail pipe, the lower end of which is immersed in water. The injection water is forced through the nozzles by the pressure and the existing vacuum The jets are directed into the throat of the combining tube where they unite to form a single jet. The exhaust steam enters the condenser through the steam inlet, comes into direct contact with the converging water jets, and is condensed.

From the combined effect of the external water pressure, the vacuum existing in the condenser, and gravity, the water jets attain a velocity sufficient to entrain the condensed steam, air, and noncondensable gases and to discharge them into the hot well against the pressure of the atmosphere. The water jets create a vacuum by condensing the steam and maintain the vacuum by entraining and removing the air and noncondensable gases. No air pump or water-removal pump is required. The connection shown between the turbine and the condenser consists of a corrugated copper tube with cast-iron flanges. This joint permits expansion and contraction caused by temperature changes.

The ejector condenser requires more water than other forms of jet condensers but the absence of vacuum pumps may offset this disadvantage. Multijet condensers can maintain vacua up to 29 in., referred to a 30-in. barometer, with 60 F cooling water; this type of condenser is suitable for turbine installations up to 10,000-kw capacity. In the low-level jet condenser the condensing chamber is at a low elevation, water being removed by a pump, and the over-all height of the unit is low enough so that the condenser may be directly beneath the turbine or engine. In the barometric type, the condenser is placed at a sufficiently high level (about 40 ft) above the point of water discharge to enable the water to drain away by gravity through a water-sealed tail pipe or barometric column.

Figure 178 shows a low-level jet condenser of the parallel-flow design; i.e., the steam, cooling water, and noncondensable gases flow in the same direction. The inlet cooling water is drawn into an annular chamber at the top by the vacuum within the condenser and flows into the condenser chamber through a number of helical distributing nozzles which reduce the water to a spray, thus insuring direct contact with the incoming steam. The mixture of cooling water and condensate passes through a cone in which the air is entrapped in the water. The mixture then falls in a stream through the lower part of the condenser body into the pump opening and is removed by a centrifugal pump working against the vauum in the condenser.

The cooled air-vapor mixture is withdrawn near the top of the condenser by an air pump or air ejector. In order to prevent the water from rising to an excessively high level in the condenser, a vacuum breaker is provided. This breaker consists of a float-controlled valve. If the pump should fail and the water level rise, the valve would open and admit air. An atmospheric relief valve is connected to the top of the condenser. This valve opens automatically when the pressure in the condenser becomes greater than that of the atmosphere and closes when there is a partial vacuum in the condenser. When the atmospheric

exhaust valve is open, the turbine exhaust steam flows to the atmosphere.

Figure 185 shows the front and sectional views of a low-level jet condenser. The wet-vacuum pump removes both the condensate and the entrained air and other noncondensable gases. The liquid con-

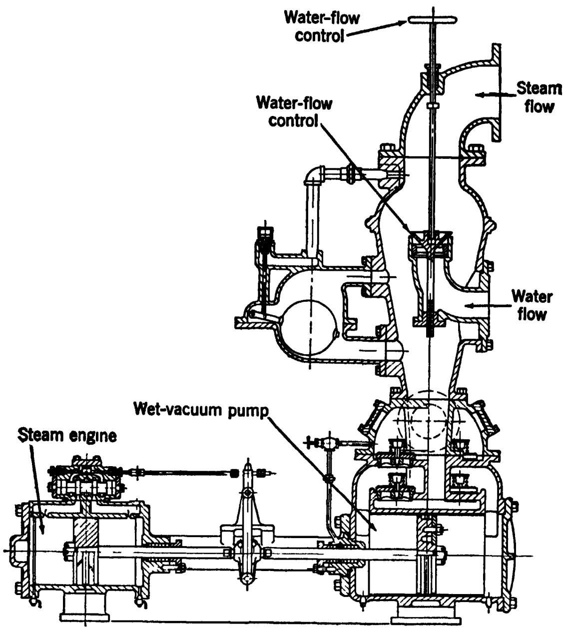


FIG. 185. Worthington jet condenser with wet-vacuum pump.

densate helps to seal the piston rings against leakage; no separate air ejector is necessary. These condensers are made of cast iron and of bronze, the latter being used for salt water injection. Condensers of this type are made in sizes capable of condensing from 1700 to 25,500 lb of steam per hr when cooled with 70 F condensing water and operating against 4 in. mercury absolute exhaust pressure.

The adjustable cone admits water in thin conical sheets at the end of the injection elbow. Adjustments can be made in water flow to meet

variations in steam load and cooling water temperature. A vacuum breaker in the form of a copper ball float prevents the water from rising

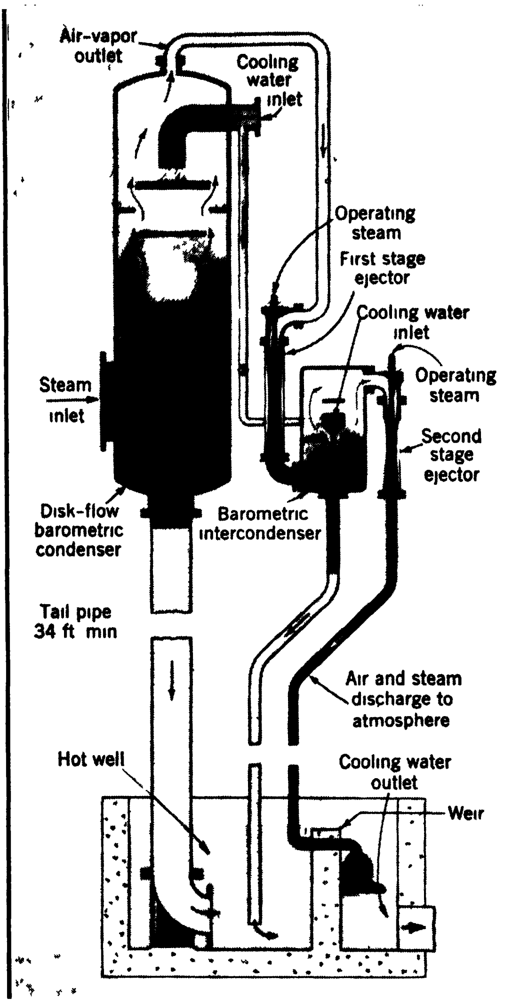


FIG. 186. Ingersoll-Rand disk-flow counter-current (direct-contact) barometric condenser.

in the condenser if the pump should fail. A simplex steam engine drives the wet-vacuum pump.

228. Barometric Condensers. Figure 186 shows a section through a countercurrent (direct-contact) barometric condenser, incorporating the disk-flow method of water distribution. In the condenser illus-

trated, the cooling water enters at a point above the steam inlet, and the water drops from disk to disk, as shown. The air content is removed by a two-stage steam-jet air ejector with intercooling. High-pressure steam expanding through nozzles at high velocity entrains the air and noncondensable gases; the kinetic energy of this high velocity is converted into pressure in the combining throat to compress and force out the steam-air-vapor mixture.

The hot water resulting from the condensing process falls to the bottom of the condenser, then into the tail pipe, and the air at the extreme top is cooled to a temperature near that of the inlet water. Thus the air ejector handles cool gases containing little vapor and practically no water. The lower end of the tail pipe (about 35 ft in length) dips into the hot well. As atmospheric pressure can support a column of water about 34 ft high, the tail pipe forms an automatic removal pump, and water flows from the tail pipe as rapidly as it accumulates.

In both low-level and barometric condensers it is usual practice to lift the water from the source of supply to the injection head by means of the vacuum within the condenser. The greatest height to which the water can be lifted by this means is about 18 ft with a 28-in vacuum, referred to a 30-in. barometer. When necessary, a pump is used to aid in lifting the water to the condenser head. The construction of the barometric condenser is simple—no moving parts, no nozzles, no contracted orifices to become clogged, and no need for vacuum breakers—but a long exhaust pipe is required.

229. Vacuum Pumps. The vacuum or air pumps used with steam-condensing plants are of four types: *reciprocating-piston*, *positive rotary-displacement*, *hydrocentrifugal*, and *steam-jet*. The earliest dry-vacuum pumps were merely air compressors of the piston type which worked between a high vacuum and atmospheric pressure. These pumps receive saturated "air" at the pressure existing in the condenser and compress it to a pressure high enough to discharge it to the atmosphere. Some reciprocating pumps are arranged with a crank and flywheel and are known as *rotative dry-vacuum pumps*. Leakage, defective jacket cooling, and re-expansion of the air in the clearance spaces make it difficult to maintain a vacuum of more than 26 in. of mercury referred to a 30-in. barometer. Another disadvantage of the reciprocating air pump is the excessive size and the high cost when the pump is made for large capacity. Positive rotary-displacement vacuum pumps are usually of vane-type construction.

In the hydraulic air pump, also called hurling-water pump, a rapidly rotating impeller continuously hurls thin sheets of water through suitable water passages into a Venturi-shaped discharge cone where the high-velocity water stream entrains the air from the condenser.

A volume of air is compressed between two successive sheets of water and carried into the discharge cone. This mixture of air and water then passes into an expanding diffuser cone, where the energy of the high-velocity stream is converted into pressure to discharge the water and entrained air into the atmosphere. The water is used again with an

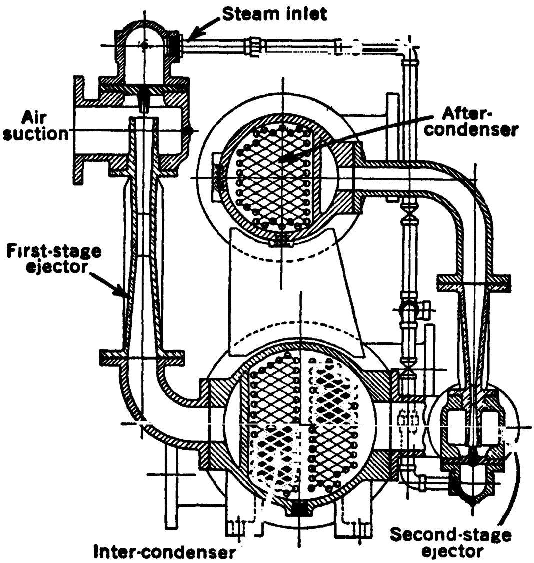


FIG. 187. Section of Elliott two-stage steam-jet air pump.

addition of about 10 per cent make-up to reduce the temperature since very little heat is abstracted from the air-vapor mixture. The hydraulic entrainment type of air pump is used with larger condenser installations than the reciprocating-piston type chiefly because of its compactness, high air-removal capacity, and ability to maintain a slightly higher vacuum. However, the hydraulic air pump requires more power than the reciprocating pump.

The *steam-jet air-ejector* type of vacuum pump has practically supplanted other types for steam condensers when high vacua are required and when large volumes of gases must be handled, because of its compactness, absence of moving parts, simplicity of operation, and high efficiency. One or more steam jets moving at a high velocity (about 3000 fps) entrain the saturated noncondensable gases, and the mixture passes through an expanding tube in which the velocity is reduced with a resulting increase in pressure. The capacity varies with the steam pressure and the size and number of steam nozzles.

Typical American practice is illustrated in Fig. 187, which shows a

two-stage ejector with a combined inter- and aftercondenser of the surface type. In this equipment, high-pressure steam flows through the first-stage ejector entraining air and vapor from the main condenser. This mixture flows into the intercondenser where the steam is condensed and the air and gases are cooled prior to being picked up by the second-stage ejector. The condensate forming in the intercondenser at approximately 7 psia is usually drained through a trap back to the main condenser.

Live steam is admitted to the second-stage ejector which draws the air and gases from the intercondenser and discharges them into the aftercondenser. The entering steam is condensed and usually drained to an open feedwater heater. The air and gases float out to the atmosphere.

230. Heat Absorbed by a Condenser. The size of the condensing equipment depends on the total enthalpy of the entering steam and on the quantity and temperature of the cooling water. When the condenser is bolted directly to a simple nonreheating and nonextracting steam turbine- or engine-generator set, the enthalpy supplied to the condenser is that supplied to the prime mover, exhausting to the condenser, less that converted into work and consumed by generator losses, friction, convection, radiation, and other losses not readily determined. Hence

$$m_s h_t = m_s h_x + \frac{3413K}{e_m} \tag{227}$$

or

$$h_x = h_t - \frac{3413K}{e_m m_s} \tag{228}$$

where h_x = enthalpy of steam at exhaust, Btu per lb.
h_t = enthalpy of steam at throttle, Btu per lb.
3413 = equivalent of 1 kilowatt-hour, Btu per kw-hr.
K = load on prime mover, kw.
m_s = steam supplied to prime mover, lb per hr.
e_m = combined mechanical and electrical efficiency of prime mover, expressed decimally, including all friction, electrical loss, windage external to the turbine, convection, and radiation.

When the load K on the prime mover is given in horsepower, the constant 2545 must be substituted for 3413. The value of e_m varies with both the type and size of prime mover and may be assumed from general practice or obtained from the manufacturer.

231. Cooling Water for Jet Condensers. In jet condensers the cooling water and condensate are discharged as a mixture and therefore have the same average discharge temperature. Neglecting the effect of convection, radiation, leakage, and entrained air, the heat given up by the exhaust steam is equal to the heat received by the cooling water. Hence

$$m_s(h_x - h_{f,2}) = m_w(h_{f,2} - h_{f,1})$$

or

$$m_w = \frac{m_s(h_x - h_{f,2})}{(h_{f,2} - h_{f,1})} \tag{229}$$

where m_w = weight of injection water required, lb per hr.
$h_{f,2}$ = enthalpy of mixture leaving condenser, Btu per lb.
$h_{f,1}$ = enthalpy of injection water entering condenser, Btu per lb.
Other items are the same as for equation 228.

The temperature of the mixed cooling water and condensate leaving the condenser would be the same as that corresponding to the absolute pressure in the condenser if it were not for the partial pressure (Dalton's law) exerted by the noncondensable gases. In high-vacuum jet condensers the air pressure is low and the difference ranges from 6 to 10 F; in low-vacuum jet condensers the difference may be 12 to 20 F. The weight of cooling water required per pound of steam will vary from 20 to 60 lb, depending mainly on the initial temperature of the water and the vacuum desired. Experience has shown that 10 per cent or more should be added to the calculated values of m_w from equation 229 to obtain the weight of injection water which would probably be used.

232. Circulating Water Required for Surface Condensers. Attempts have been made to dispose of the heat rejected in a surface condenser by passing air over the cooling surfaces. The power consumption involved in attaining the necessary cooling effect rules the air-cooling method out of practical consideration except in locomotive practice. The only effective cooling medium is water. The weight of cooling water varies with the design of the condenser and the conditions of operation, such as diameter, thickness and spacing of the tubes, cleanliness of tube surfaces, initial temperature and velocity of the circulating water, and vacuum desired. The quantity of circulating water required can be determined approximately by calculation, and the values thus found must then be increased by such factors as experience has shown to be necessary.

For surface condensers, the heat given up by the exhaust steam equals the heat absorbed by the circulating water, neglecting the effect of con-

vection, radiation, leakage, and entrained air.

$$m_s(h_x - h_{f,c}) = m_w(h_{f,2} - h_{f,1})$$

or

$$m_w = \frac{m_s(h_x - h_{f,c})}{(h_{f,2} - h_{f,1})} \tag{230}$$

where m_w = weight of circulating water required, lb per hr.
$h_{f,c}$ = enthalpy of condensate leaving condenser, Btu per lb.
$h_{f,2}$ = enthalpy of circulating water leaving condenser, Btu per lb.
$h_{f,1}$ = enthalpy of circulating water entering condenser, Btu per lb.

Other items are the same as for equation 228.

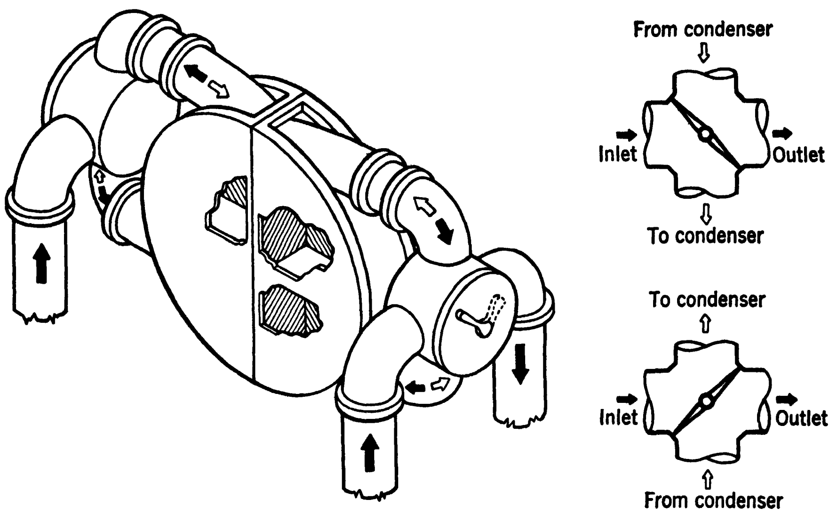


FIG. 188. C. H. Wheeler flow-reversal method for surface condensers.

The discharge temperature of the circulating water is generally 10 to 20 F less than the temperature corresponding to the steam pressure in the condenser. The temperature of the condensate is often 15 or more degrees below that of the exhaust steam, but good design allows the condensate temperature to approximate very closely the temperature corresponding to the total pressure in the condenser. The weight of circulating water required per pound of steam will vary from 25 to 50 lb with low vacua and 60 to 120 lb for vacua over 27 in. of mercury, referred to a 30-in. barometer. An average figure for steam-turbine condensers using high vacua is 8 lb of steam condensed per hr per sq ft of tube surface. In some central-station installations, surface condens-

ers of the vertical single-pass type are employed; the surface varies from 0.65 to 1.15 sq ft per kw of turbine rating. For the horizontal single- and two-pass types, about 0.90 to 1.75 sq ft per kw are required. Small turbine generators require from 2 to 4 sq ft of tube surface per kilowatt.

One scheme for improving the mean coefficient of heat transmission and reducing the time lost for cleaning is shown in Fig. 188. By periodic reversal of the water flow valves the inlet tube sheet can be kept free of the accumulations of refuse normally carried by the water—leaves, grass, seaweed, and marine growth.

233. Heat Transfer in Surface Condensers. The performance of a surface condenser is expressed in Btu transmitted per square foot of tube surface, per hour, per degree mean difference in temperature between the fluids on the two sides of the tube. Neglecting external convection, radiation, and leakage, the heat absorbed by the circulating water is equal to the heat given up by the exhaust steam, or

$$Q = UA\theta_m = m_s(h_x - h_{f,c}) \tag{231}$$

$$A = \frac{Q}{U\theta_m} = \frac{m_s(h_x - h_{f,c})}{U\theta_m} \tag{232}$$

where Q = heat transferred, Btu per hr.
A = area of cooling surface, on steam side, sq ft.
U = mean coefficient of heat transmission, Btu per (hr) (sq ft) (deg F mean temperature difference).
θ_m = logarithmic mean temperature difference between the steam and circulating water, deg F; see equations 233 and 234.
Other items are the same as in equations 228 and 230.

The coefficient U, as used in equation 232, refers to the mean or average value for the entire tube surface and not the local value since that varies widely for different parts of the condenser. The local value varies from more than 1000 Btu for air-free steam in the upper part of the condenser to about 5 Btu for tubes surrounded only by air (as approximated in the lower part of a surface condenser). The results of tests on various makes of surface condensers formed the basis for the coefficients of heat transfer indicated by Fig. 189. These curves show that the coefficient increases with an increase in water velocity through the tubes; that it varies with the inlet-water temperature, being greater with high than with low water temperatures; and that tubes of small diameter give a somewhat higher coefficient of transfer than those of larger diameter. It is also known that the condition of the tube surfaces (inside and out), with respect to cleanliness, air film, and scale,

largely influences the heat transmission. High vacua in condensers are primarily dependent on the lowest possible inlet cooling-water temperature.

Figure 189 shows heat-transfer curves for three tube sizes, namely, ¾-in., ⅞-in., and 1-in. O.D. (outside diameter) tubes, each with a thickness of No. 18 B.W.G.; the ¾-in. tube has the highest rate of transmission. The curves are based on an inlet circulating-water tem-

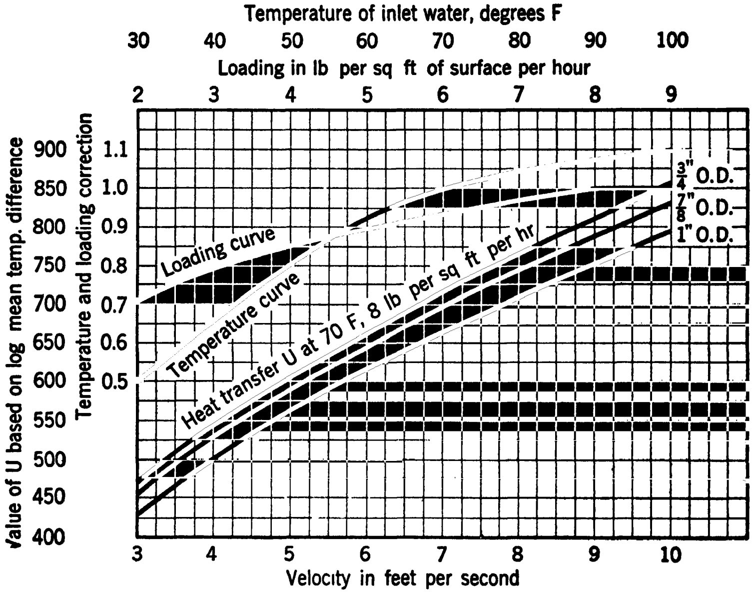


FIG. 189. Heat-transfer curves for surface condensers serving steam turbines; for condensers serving engines use 0.65 of above values of *U*.

perature of 70 F and a steam supply of 8 lb per sq ft per hr. A temperature correction curve is shown for other water temperatures. A loading correction curve is shown for partial load conditions. The entire cooling surface, although available, may not be effectively used, and the mean value of *U* is reduced. If a condenser were designed for 70 F inlet water, 8 lb of steam per sq ft per hr, ¾-in. tubes, and a velocity of the circulating water through the tubes of 6 fps, the value of *U* from Fig. 189 would be 660. If this condenser were supplied with 60 F water at a velocity of 4.5 fps through the tubes and 4 lb of steam per hr per sq ft, the value of *U* would be $0.92 \times 0.85 \times 575 = 450$. The above values assume commercially clean tubes.

The coefficient *U* is multiplied by a condition factor of 0.8 to 0.9 to

account for the difference between clean and dirty tubes caused by normal water conditions. For engines with oily exhaust this factor is usually taken as 0.65. The quantity of heat passing through the cooling surface of any heat exchanger is proportional to the temperature difference at any instant for a given point. The local temperature

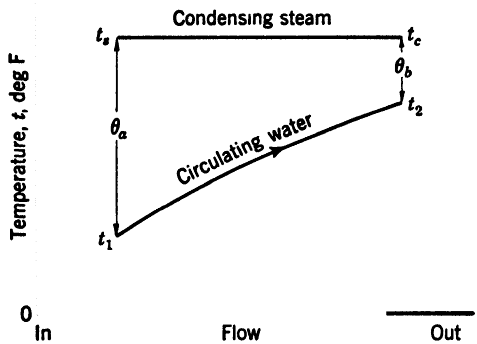


FIG. 190. Temperature difference in condensers.

difference is indeterminate; hence it is necessary to establish a mean temperature difference for the entire period of thermal contact of steam and circulating water. The logarithmic or the arithmetical mean temperature may be taken for this purpose, but the determination of either value is based upon assumptions which do not represent actual conditions. In general, from Fig. 190,

$$\theta_m = (\theta_a - \theta_b) \div \log_e \frac{\theta_a}{\theta_b} \tag{233}$$

where θ_m = the logarithmic mean temperature difference, deg F.
θ_a = difference in temperature between incoming hot fluid and its surroundings, deg F.
θ_b = difference in temperature between outgoing hot fluid and its surroundings, deg F.

Also, from Fig. 190,

$$\theta = \left(\frac{t_s + t_c}{2}\right) - \left(\frac{t_2 + t_1}{2}\right) = \frac{(t_s - t_2) + (t_c - t_1)}{2} \tag{234}$$

where θ = arithmetical mean temperature difference, deg F.
t_s = temperature in steam space, deg F.
t_c = temperature of condensate leaving condenser, deg F.
t_2 = temperature of circulating water leaving condenser, deg F.
t_1 = temperature of circulating water entering condenser, deg F.

Most engineers prefer the logarithmic mean, which should be adhered to for all heat-transfer calculations where the range of temperature is high, because the assumptions more nearly approach actual conditions. However, the arithmetical mean is sufficiently accurate for purposes of comparison, especially when the rise in temperature of the circulating water is less than 10 deg F.

234. Water-Cooling Methods. When an ample supply of water is unobtainable at a low cost or from a natural source, some means of cooling the circulating water must be found. The cooling water is often circulated continuously, being cooled after each passage through

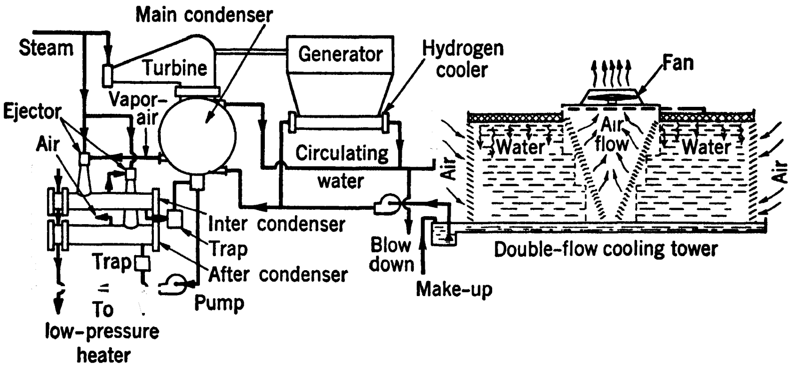


FIG. 191. Flow diagram for a surface condenser.

the condenser. The cooling is effected by the exposure of a large surface of the water to the air. The resultant evaporation of some of the water with the absorption of the latent heat of vaporization cools the remainder so that the cooled water can be circulated again. This process of recooling the water is usually accomplished by means of cooling ponds or cooling towers.

The oldest means of cooling and storing condenser circulating water is to discharge the hot water into a pond of sufficient area so that the water is cooled by the air passing over the surface of the pond. In order to reduce the large area necessary, the heated water may be sprayed into the air over the pond, thus increasing the rate of cooling. The cooling action is effected by convection and evaporation. Natural ponds without spray nozzles require 30 to 50 times the area of spray cooling systems.

Because of space requirements and windage loss with the spray pond, the cooling tower has been developed. Cooling towers are rectangular wood or metal structures generally arranged with some form of baffling

devices. The hot water is delivered to the top and falls in thin sheets or sprays into a tank below. Air enters at the bottom and flows upward, cooling the water by contact (convection) and by the partial evaporation which results. The circulation of air may be effected by either natural or forced draft. With natural draft—the difference of temperature between the air inside and outside the tower causing the flow—the chimney action may be aided by fitting a stack to the top of the tower. Forced-draft towers have closed sides, except for openings containing one or more fans; the heated and saturated air leaves through the open top of the tower.

Figure 191 shows a typical flow diagram for a surface-condenser cooling-water system. The cooling load consists of the main condenser and the hydrogen cooler for the generator as well as an oil cooler not shown.

PROBLEMS

1. Determine the theoretical amount of 70 F cooling water needed per pound of dry saturated steam entering a jet condenser which is operating at a pressure of 1.47 psia.

2. A barometric condenser serves a steam turbine which is exhausting 6000 lb of steam per hr at a pressure of 0.98 psia and having a quality of 90 per cent. (*a*) Determine the gallons per minute of 60 F cooling water needed. (*b*) Determine the temperature of the cooling water at the outlet.

3. If a surface condenser is used instead of the barometric condenser in problem 2, determine (*a*) the energy, Btu per hour above 32 F, that would be recovered in the condensate if no subcooling occurred; and (*b*) the cubic feet per minute of circulating water required at 60 F inlet and 80 F outlet temperatures.

4. A steam engine delivers 100 bhp when supplied with steam at 150 psia and 98 per cent quality. It exhausts into a jet condenser at a pressure of 4 in. mercury absolute. Find the pounds of 60 F injection water theoretically required per hour if the engine uses 28 lb of steam per bhp-hr, and the mechanical efficiency is 90 per cent. Neglect radiation. 42,200 lb per hr, *Ans.*

5. Determine the tube area required for a condenser operating under the following conditions: steam inlet temperature, 110 F; circulating-water temperature at inlet, 70 F and at outlet, 98 F; condensate temperature, 110 F. The over-all coefficient of heat transfer U is 700 Btu per (hr) (sq ft) (deg F). Steam flow is 50,000 lb per hr. Enthalpy of entering steam is 1012 Btu per lb.

6. Using the data of problem 5, determine the number and length of ¾-in. O.D. tubes having a wall thickness of 0.049 in. necessary for a two-pass condenser if the water velocity is to be 7 fps. 460 tubes per pass, 15.83 ft, *Ans.*

7. A surface condenser with a cooling surface of 2630 sq ft is supplied with 3475 gal per min of cooling water which enters at 32 F and leaves at 54 F. The condensate leaves the condenser at 91.72 F, and the pressure in the condenser is 1.5 in. mercury absolute. (Weight of water is 8.33 lb per gal.) Find (*a*) the heat absorbed by the cooling water, Btu per hour; (*b*) log mean-temperature difference between steam and cooling water, degrees F; and (*c*) coefficient of heat transfer U, **Btu per** (hr) (sq ft) (deg F).

8. A surface condenser receives exhaust steam having a quality of 90 per cent. The vacuum in the condenser is 27.5 in. mercury referred to a barometric reading of 29.5 in. mercury. Cooling water enters at 60 F, leaves at 80 F, and the temperature of the condensate is 85 F. How many pounds of cooling water are required per pound of steam condensed?

9. Steam enters a surface condenser at a pressure of 1 in. mercury absolute, with a velocity of 600 fps, and a quality of 95 per cent, and leaves as condensate at a temperature of 75 F and with a velocity of 10 fps. Cooling water enters at 60 F and leaves at 70 F. Determine (*a*) the pounds of cooling water required per pound of steam condensed; (*b*) the total gallons of cooling water required per minute for a 25,000-kw turbine with a steam rate of 12 lb per kw-hr; and (*c*) the diameter of the entrance to the condenser.

10. A 10,000-kw turbine generator uses 12.8 lb of steam per kw-hr at rated load. Initial steam pressure is 500 psia; temperature, 600 F; back pressure, 1 in. of mercury absolute. Cooling water enters at 65 F and leaves at 75 F, and the condensate has a temperature 4 F less than exhaust steam temperature. The efficiency of the turbine generator is 92 per cent (combined electrical and mechanical efficiency of prime mover and generator includes friction, windage, and other losses). (*a*) What weight of circulating water is required per hour? (*b*) If U, based on the log mean-temperature difference, is 810 Btu per (hr) (sq ft) (deg F), how many square feet of condenser tube surface are required?

11. In a surface condenser, the initial temperature of the circulating water is 55 F and the exit temperature is 100 F. Dry saturated steam enters the engine at 140 psia, and the engine exhausts into the condenser at 2 psia. The engine develops 180 ihp and uses 20 lb of steam per ihp. The final temperature of the condensed steam is 126 F. (*a*) What is the quality of the steam entering the condenser? (*b*) What is the thermal efficiency of the engine based on the indicated horsepower?

(*a*) 95 per cent, (*b*) 11.59 per cent, *Ans.*

12. Using the heat-transfer curves of Fig. 189, find the value of U for a surface condenser using ¾-in. O.D. tubes and having a loading factor of 8 lb of steam per sq ft, a cooling water temperature of 70 F, and a velocity of 8.75 fps.

$U = 800$, *Ans.*

13. Using the heat-transfer curves of Fig. 189, find the value of U for a surface condenser using 1-in. O.D. tubes having a loading factor of 4 lb of steam per sq ft, a cooling water temperature of 50 F, and a velocity of 9.5 fps.

14. Calculate the square feet of tube surface required for a surface condenser receiving steam from a turbine whose steam rate is 18 lb per kw-hr. Steam is supplied to the turbine at 200 psia and 480 F, and the condenser pressure is 2 in. mercury absolute. Load on generator, 3000 kw; mechanical efficiency of generator, 90 per cent. Assume no losses, and no subcooling of condensate. Water temperature at inlet is 75 F and at outlet, 95 F. Use $U = 650$. 5890 sq ft, *Ans.*

15. A 2000-kw steam turbine uses 12 lb of steam per kw-hr at rated load when supplied with steam at 600 psia and 700 F total temperature and with a condenser pressure of 2 in. of mercury absolute. Calculate the square feet of ¾-in. tube surface required in a surface condenser for this unit; inlet water at 85 F; outlet water at 95 F; no subcooling of condensate. Assume no losses and water velocity through the tubes at 7 fps. Use Fig. 189 to find U. Assume e_{me} as 0.92. (This problem requires successive approximations)

16. A 10,000-kw turbine generator uses 10 lb of steam per kw-hr at rated load. Steam supply pressure, 650 psia; temperature, 700 F; and the vacuum in the

surface condenser, 29 in., referred to a 30-in. barometer. Temperature of inlet circulating water, 60 F; outlet water, 72 F; and no subcooling of condensate. Mechanical efficiency of the generator is 92 per cent. The condenser tubes are ¾-in. O.D.; thickness, 0.049 in.; water velocity in tubes, 7 fps. Assuming no losses, find (*a*) gallons of circulating water required per minute; (*b*) condenser tube surface, square feet; (*c*) number of tubes for a two-pass condenser, and (*d*) length of tubes in feet. Determine *U* from Fig. 189.

CHAPTER

13

Pumps

235. Classification. Pumps, for imparting velocity and pressure to fluids, may be grouped as follows: (1) piston units with one or two cylinders, and one-, two-, or three-cylinder plunger pumps; (2) single and multistage centrifugal machines; (3) rotary devices with screw impellers, vane devices with either swinging or guided parts, radial plungers, eccentric pistons, lobar impellers, and internal-gear units with equal or unequal numbers of teeth in the rotors; and (4) fluid-impellent pumps with either continuous or intermittent flow.

Piston pumps work by exerting a pressure directly on the fluid being pumped. Centrifugal pumps exert the pressure indirectly by gradually arresting the momentum of the fluid given to it by rapid rotation of an impeller within a suitable casing. The water inlet and discharge of the piston pump are controlled by valves which open and close intermittently, whereas in the centrifugal pump the entry and discharge are continuous with no valves and no pump controlling devices. Rotary pumps, similar to Fig. 203, combine the positive-pressure method of the piston pump with the continuous entry and discharge of the centrifugal pump by employing rotating impellers, screws, gears, oscillating pistons, and rotors in connection with either stationary or guided vanes. They are generally of light weight, low first cost, valveless, with capacities ranging from small to large, and may be employed to give discharge pressures up to 1000 psig with light to heavy oils. Their usual range of working pressures falls between 100 to 500 psig. Rotary pumps find considerable usage in the handling of heavy viscous fluids.

In fluid-impellent pumps the velocity and pressure of one fluid act directly on another, thereby imparting all or part of its energy to the latter fluid. Injectors, ejectors, aspirators, and some air lifts are of the continuous-flow type.

236. Direct-Acting Steam-Driven Piston Pumps. A piston or reciprocating pump in its simplest form consists of a cylinder provided with inlet and outlet openings, suitably controlled by valves, and a reciprocating piston or plunger working within the cylinder. This type of

pump may be used for forcing feedwater into a boiler, for removal of condensate and air from a condenser, and for hydraulic services in elevators, lifts, and presses. Direct-acting steam pumps have two forms of design: *duplex* and *simplex* units.

Duplex pumps have two steam and two water cylinders with the steam cylinders placed side by side as shown in the plan view of Fig. 192. Likewise the water cylinders are adjacent to each other at the opposite end of the assembled unit. All steam and water cylinders are double

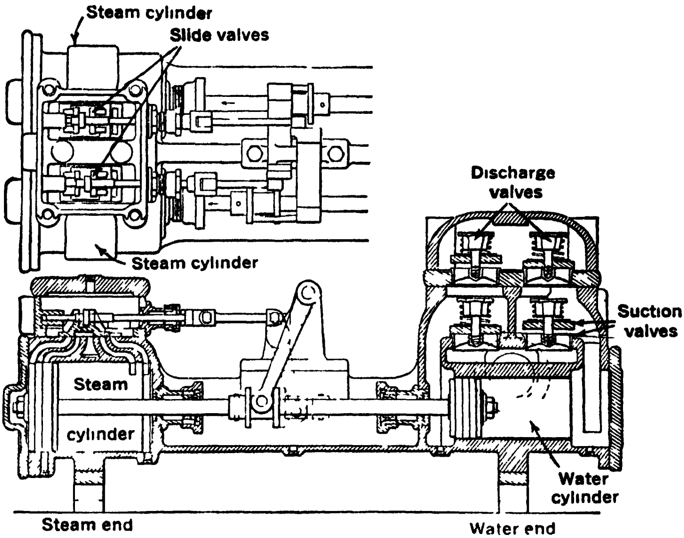


FIG 192 Plan and sectional elevation of duplex double-acting steam-driven pump.

acting in that work is done in both ends of each. In the case of all direct-acting steam pumps a piston in a steam cylinder is mounted upon the same piston rod that carries the piston of the water cylinder served by it. The water piston of Fig 192 has two circular metal plates with fibrous disks held between them. The latter serve to prevent leakage of the fluid handled between the piston and the cylinder walls This arrangement known as piston-packed reduces the leakage from one cylinder end to the other when the pump is in operation. Each end of the steam cylinders has separate steam ports for live and exhaust steam as shown by Fig. 192. This arrangement reduces the necessary amount of steam-valve travel and reduces leakage of the steam by the valve. The valve gear for a steam cylinder is actuated by the piston rod of the adjacent cylinder of a duplex pump. Direct-acting steam pumps are not economical in the use of steam, as they do not expand

the steam since steam cut-off comes at the end of the stroke. Figure 192 indicates that the steam-cylinder slide valve is giving unrestricted admission of steam to the crank end of the steam cylinder. The exhaust-steam port in the head end of the cylinder is closed; thereby some cushioning of the reciprocating parts is obtainable by steam entrapped in the head-end portion of the steam cylinder.

A simplex pump has a single steam and a single water cylinder with their pistons mounted at the ends of a common piston rod. The steam and water valves may be similar to those of Fig. 192, and the single piston rod of a simplex pump often actuates its steam-valve gear.

The pump valves of Fig. 192 consist of rubber-composition disks supported by brass caps and held firmly to their flat seats by coil springs. For hot water and pressures above 200 psig spring-loaded metallic valves may be used. The function of the types of pumps under discussion is to take a fluid such as water from some level and to deliver it under pressure to some other level. Figure 192 indicates how the water piston and water-cylinder valves function to accomplish the desired action. When the water piston moves to the left, fluid is forced through the opening beneath the lifted discharge valve (upper left), as indicated. At the same time a partial vacuum is produced in the cylinder to the right of the piston, and, if the pressure reduction is sufficient, water or other fluids will flow through the suction pipe (connection shown by dotted lines), the suction valve (lower right) will lift, and the head end (right end) will fill with the fluid being handled. When the piston moves to the right, discharge will take place under the lifted discharge valve (upper right), and flow into the left end of the fluid cylinder will be through the lifted suction valve (lower left). The water or fluid cylinder of the simplex pump functions in the same manner. Fluids may be lifted under atmospheric pressure exerted upon their surface, or they may flow under pressure greater than atmospheric pressure to the water cylinder. The height that a fluid may be lifted by suction is dependent upon its temperature and the friction and other losses within the supply connection to the hydraulic cylinder.

The sizes of direct steam pumps are stated by the diameters of the steam and the water cylinders and the length of the stroke. The specification of either simplex or duplex must be made. The statement 10 in. by 6 in. by 12 in. specifies a steam-cylinder dimension of 10-in. diameter, a water-cylinder dimension of 6-in. diameter and a rated stroke of 12 in. Direct-acting steam pumps do not have either a crank or a flywheel; therefore the length of their strokes may be equal to, less than, or greater than the rated length. The two latter conditions are termed understroking or overstroking. Both duplex and simplex

pumps have a single steam, one exhaust, one liquid-supply, and one liquid-discharge connection. Inasmuch as the piston area in the steam cylinder is always greater than that in the liquid cylinder, direct-acting steam pumps are able to discharge fluids at pressures greater than those of the steam supplied to them.

In order to give steadiness of liquid flow, air chambers are used with direct-acting steam pumps. For small pumps this chamber may be a compartment at the upper part of the cover over the discharge valves. For other units a cylindrical space closed at the upper end may be attached to either the suction pipe or the discharge pipe or both. Air entrapped in the chamber is compressed as the water piston pushes water from the cylinder, and when the stroke is completed expansion of the air in the chamber maintains a uniform rate of discharge. Air chambers are not so essential with duplex as with simplex pumps since the discharges from the liquid cylinders are not in exact phase and a steadier common-outlet flow occurs.

237. Plunger Pumps. Some form of plunger is often used in the liquid cylinder of a direct-acting pump instead of a piston. Fibrous or metal packings are placed about the plungers to prevent leakage as shown in Fig. 193. Any inside-the-cylinder piston or plunger packing is less accessible than an outside arrangement, where leakage of the liquid handled is more readily discernible. The plungers of the outside end-packed pump are moved by an external yoke which embodies two side rods which connect the plungers together. In each case the power is furnished by one or two steam cylinders located at the left of the liquid cylinder or cylinders, as in Fig. 192, depending upon whether the unit is a simplex or a duplex pump.

Piston and plunger pumps which operate at a uniform speed with the aid of the inertia effect of a flywheel are called *flywheel* or *rotative pumps*. When the reciprocating movement of the piston or plunger is derived from the rotation of a mechanically driven crankshaft, the pump is known as a *power pump*. Power pumps may be operated at constant speed by gears, belts, or chains, and the source of power can be steam, gas, or electricity. When one plunger is employed they are called simple or simplex; two plungers, duplex; and three plungers, triplex power pumps. The rotation of the crankshaft is transmitted to each plunger by a connecting rod. The base is made in compartments, and the operation is the same as for the water end of any piston pump. The efficiency of this type of pump depends largely upon the character of the driving unit and the efficiency of the transmitting mechanism. With an electric motor drive the over-all efficiency of power pumps is generally between 60 and 80 per cent.

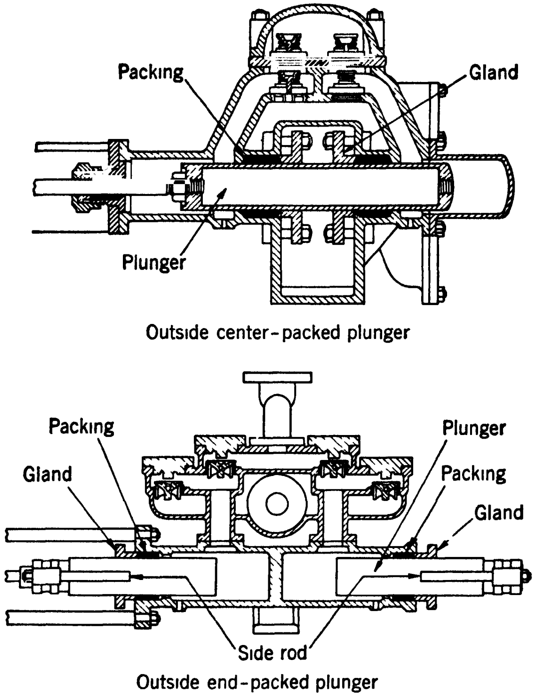


FIG. 193. Center outside-packed and outside-end-packed plunger pumps.

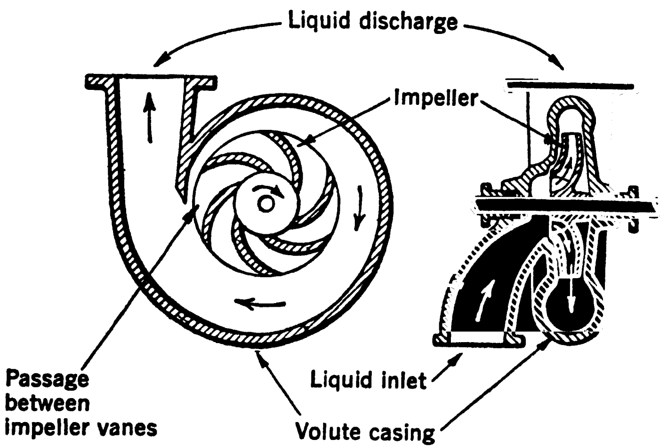


FIG. 194. Single-stage single-suction volute centrifugal pump.

238. Centrifugal Pumps. These liquid-handling machines embody the same principles as centrifugal fans used to move air and other gases. Both forms of equipment when in operation follow the same general laws. Centrifugal pumps are relatively high-speed machines and are generally direct-connected to a steam turbine or an electric motor; V-belt drives or internal-combustion engines may also be used. Water enters the impeller, Fig. 194, at the center (around the shaft), flows radially outward, and leaves the periphery of the impeller at a velocity that is the resultant of the tip velocity of the impeller blade and that of radial flow of the liquid ahead of the backward-curved impeller blade. The resultant-velocity vector is similar to that of Fig. 110, which is for a backward-curved-blade centrifugal fan. The velocity of the liquid (water) is gradually decreased, and the energy of motion is converted into pressure head in the pump casing in which the impeller rotates. The liquid being handled is under pressure and will flow out of the pump against the resistance to be overcome. The design of a centrifugal pump is generally directed towards converting the energy of velocity into potential energy without loss, minimizing the friction of impeller rotation, and balancing the end thrust of the impeller shaft.

The following groupings are applicable to commercial centrifugal pumps: *volute, turbine (diffuser), and axial-flow units; number of stages; single-suction; double-suction; open-impeller and enclosed-impeller; and horizontal and vertical.*

239. Volute Pumps. The unit of Fig. 194 has a casing of the spiral type which forms a gradually increasing water space about its impeller. In this space the liquid handled slowly loses velocity, as it flows to the discharge outlet, without the aid of diffusion vanes. The pump of Fig. 194 has a single stage, a single-liquid inlet, a horizontal shaft, a volute casing, and either an open or an enclosed impeller. The open type of impeller somewhat resembles a paddle wheel with its vanes supported at one side. Rotation of the open impeller produces excessive eddying of and leakage of the liquid handled between the blades and the casing wall, and with its single inlet produces an axial thrust on the shaft which must be counterbalanced. The open impeller gives low efficiency of operation and is used mostly with low-cost units to handle thick liquids such as mash, pulp, and sewage. The shrouded type of impeller, Fig. 195, has a plate on each end of the vanes to form enclosed water passages. The liquid handled enters the impeller at each side at its center; the enclosing shrouds or walls together with casing rings, Fig. 195, prevent it from escaping from the wheel blades.

The single-stage double-suction centrifugal pump of Fig. 196 illustrates the use of an enclosed impeller. The double-inlet impeller has

a greater liquid-handling capacity than a single-inlet rotor of a given size and operating speed. With the double-inlet impeller, axial thrusts along the line of the pump shaft are balanced without the use of either hydraulic plates or thrust bearings. A water-seal connection to each stuffing-box packing is shown in Fig. 196. Such seals help to prevent air from being drawn into the suction inlets. Air leakage into a pump suction connection reduces the height by which a liquid under atmospheric pressure may be lifted to the pump impeller inlet.

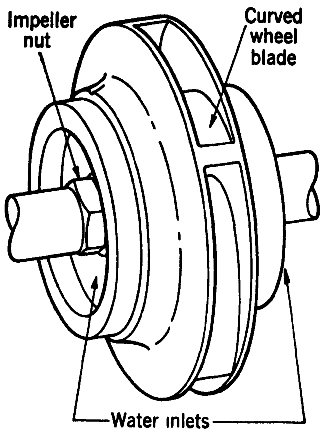


FIG. 195. Shrouded double-inlet centrifugal-pump impeller.

Impellers are usually made of some type of bronze, as it resists corrosion and maintains the original shape and smoothness of the surfaces in contact with the liquid handled. Other non-corrosive materials may be used in pump parts, depending upon the nature of the liquids handled.

240. Diffuser or Turbine Centrifugal Pumps This form of centrifugal pump employs a system of stationary diffusion vanes which sur-

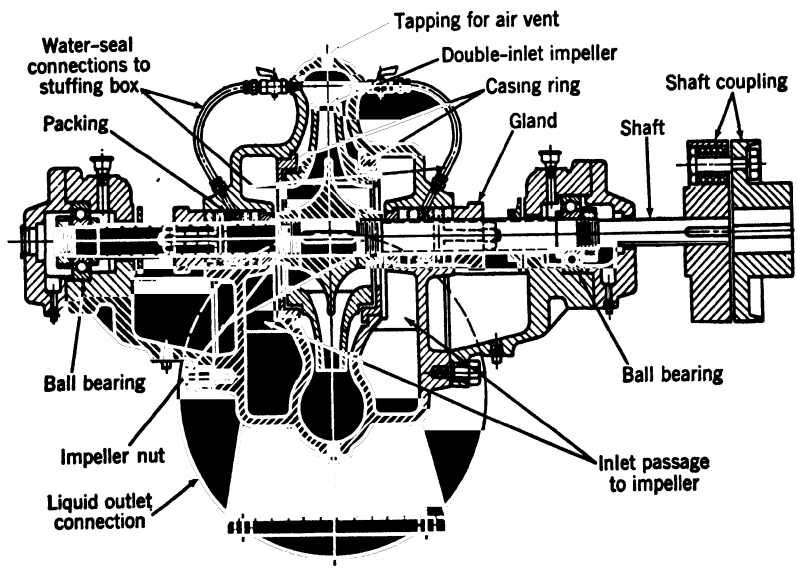


FIG. 196. Buffalo single-stage double-inlet volute centrifugal pump.

round the impeller, as shown by Fig. 197. Gradually enlarging areas are provided between the diffusers, and in these passageways the velocity of the liquid leaving the tips of the rotor blades is decreased and

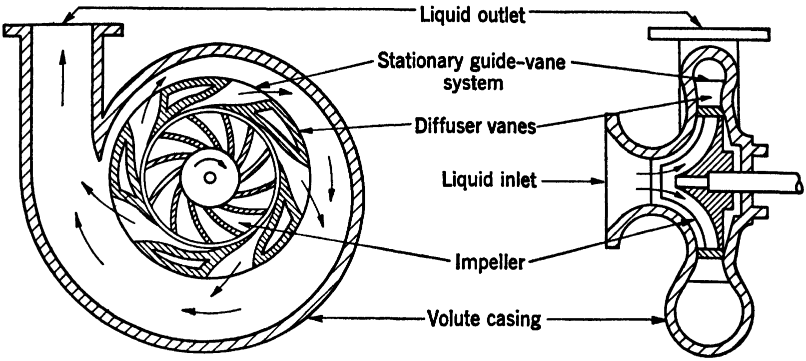


FIG. 197. Single-stage single-inlet turbine or diffuser centrifugal pump.

most of its velocity head transformed into pressure head. The velocity-pressure conversion does not take place in the casing scroll alone as in the volute form of pump. Diffuser construction may be employed in pumps of two or more stages.

241. Axial-Flow Pumps. The horizontal pumps (horizontal shaft) of Figs. 194 and 196 have impeller wheels with a shroud or shrouds essentially at right angles to the shaft axis. The same construction may be used in vertical pumps where the shaft is held in an upright position. The term radial flow is applied to the impeller blading of either of the foregoing types.

Axial-flow pumps differ from radial-flow pumps in that they have an impeller (or propeller) in which the direction of liquid flow and its forward component of velocity are parallel to the axis of the shaft upon which the rotating element is mounted as in the vertical unit of Fig. 198. Axial-flow design is also employed with horizontal pumps.

Mixed-flow pumps, Fig. 199, have impellers which discharge the material handled in a direction intermediate between those of radial- and axial-flow units. Both axial-flow and mixed-flow impellers are built with only the open-type of construction.

242. Multistage Centrifugal Pumps. The liquid involved passes through two or more impellers in multistage centrifugal pumps. Each stage consists of a separate compartment in which an impeller, mounted upon a shaft common to all stages, revolves. Each stage delivers its discharge to the inlet of the next-stage impeller with the exception of

the last stage, which sends its output to the pump outlet. The final liquid (water) pressure is dependent upon the number of stages involved as well as upon features of the pump design and its speed of operation. Commercial multistage pumps may have two, three, four, five, six, seven, eight, nine, ten, or eleven stages, depending upon the design and the builder. The casings may be either of the volute or the diffuser construction, and the impellers may be either single-inlet or the less frequently used double-inlet type.

Figure 200 illustrates a horizontal 4-stage single-suction opposed-impel-

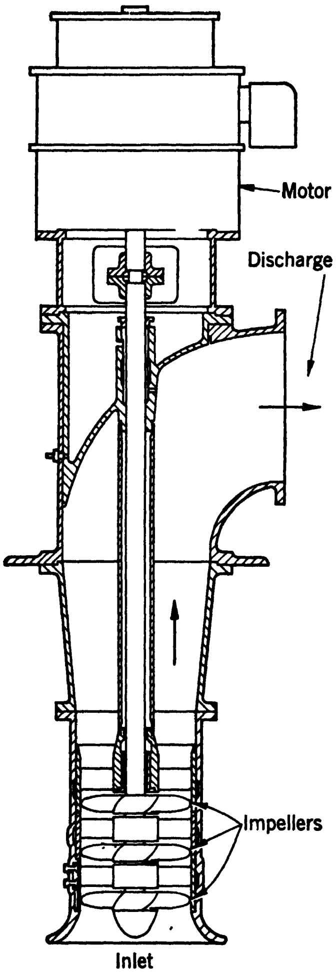


FIG. 198. Westinghouse three-stage vertical axial-flow pump.

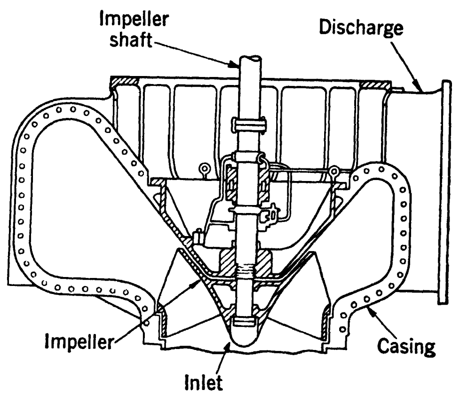


FIG. 199. DeLaval mixed-flow pump.

ler volute-casing centrifugal pump fitted with one of its bearings as a thrust-absorbing unit. The opposed arrangement of the impellers practically eliminates any axial thrust (except under unusual conditions) along the impeller shaft. When the impellers are not opposed, the pump must have a thrust bearing or a balancing drum upon which a pressure is exerted to overcome the axial thrust produced by the difference of pressure at the inlet to each impeller and that exerted on an equal area on the opposite side of the impeller.

The stages of the pump of Fig. 200 are not in sequence along the

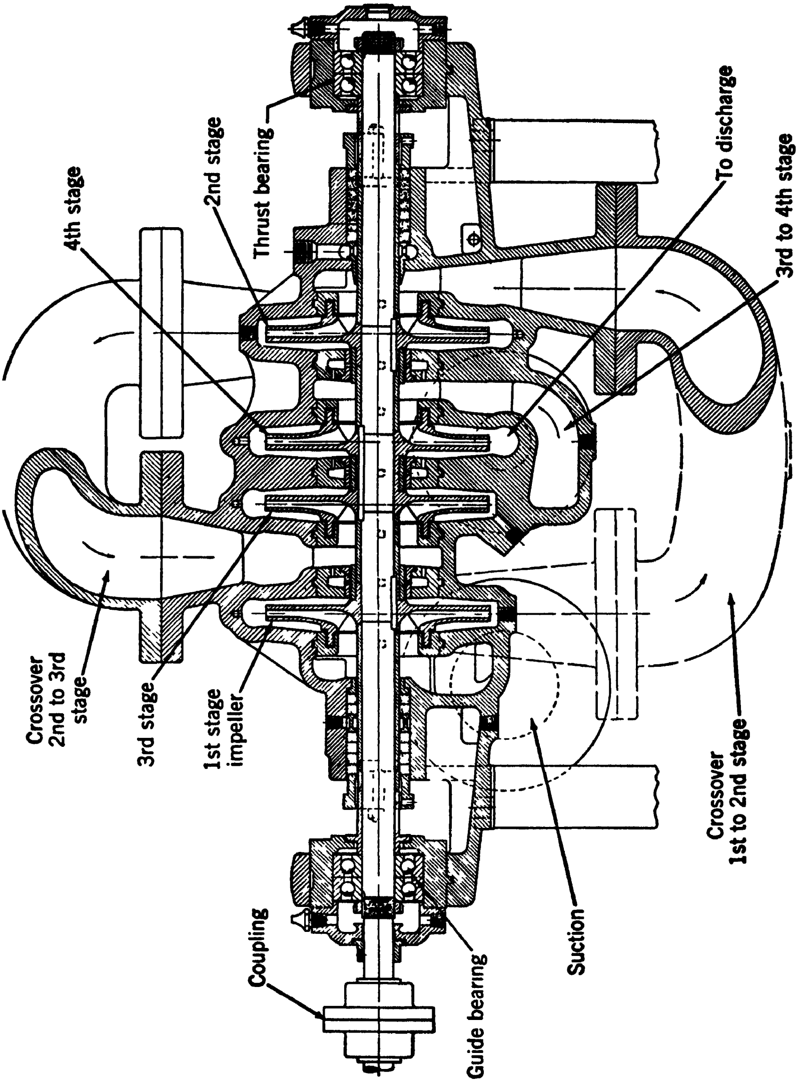


FIG. 200. Buffalo four-stage opposed-impeller volute centrifugal pump.

pump shaft. When such an arrangement of staging is made, the liquid from one stage must be conveyed to the next pressure stage by passageways which are either internal or external to the pump casing. The pump of Fig. 200 has crossover pipes between stages 1 and 2, and between stages 2 and 3. The complete connection between the discharge of stage 3 and the inlet of stage 4 is not shown.

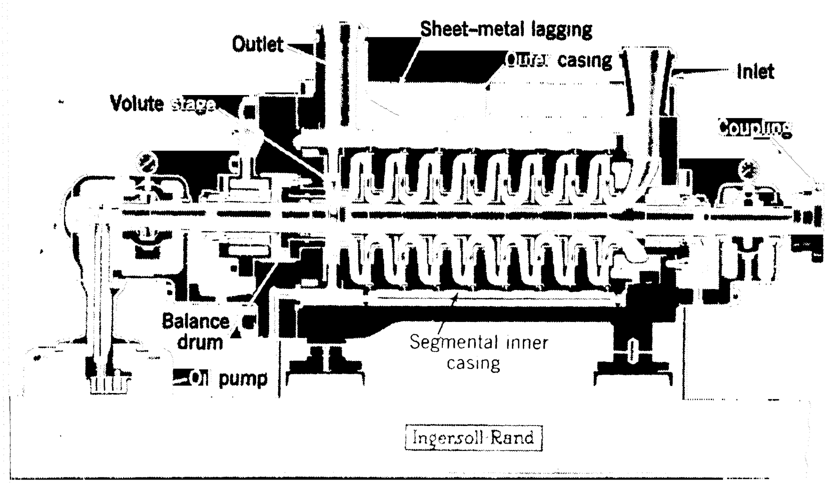


FIG. 201. Ingersoll-Rand barrel-type turbine pump.

The barrel type of multistage centrifugal pump, illustrated in Fig. 201, embodies diffuser or turbine casing construction and single-inlet enclosed impellers which operate without involved arrangements for the conveyance of the water handled as it passes from each stage. The pump inner casing is built up from separate segments, each of which forms a stage. These segments are held together rigidly, and the component parts of a stage are illustrated and named in Fig. 202. In the assembly shown in Fig. 201 the axial thrust is offset by the use of a balancing drum and a thrust bearing. All impellers are shown in section except that of the last stage which appears as an external view.

The diffusers are similar to those of Fig. 202 and with the water passages are a part of the channel ring within which the channel-ring bushing, the impeller, the channel wearing ring, the split ring, the lock ring, and the intermediate sleeve are placed. The inner casing is surrounded by water, under the discharge pressure of the last stage, held in an annular space between the inner and the outer casings. The liquid jacket thus formed permits nearly constant-temperature conditions, up to

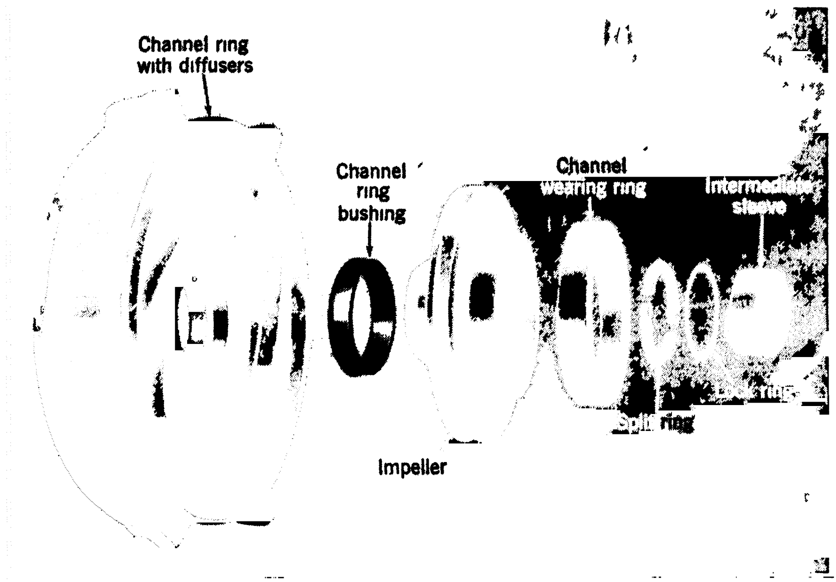


FIG 202 Ingersoll Rand barrel type centrifugal pump inner casing segment with parts

800 F, to be maintained within the pump The upper part of the assembled unit is enclosed by a sheet-metal lagging Barrel pumps are suitable for boiler-feed purposes with pressures up to 3000 to 3500 psig and high-temperature water.

243. Vertical-Shaft Centrifugal Pumps. Where conditions are such that a horizontal-shaft pump cannot be located within suction distance (less than 23 ft) of the supply water level a vertical-shaft centrifugal pump may be necessary. This condition is frequently encountered in deep wells, sewage service, sumps, and near rivers where the difference in water level varies considerably. The pump may be completely submerged in the water, but when possible it is located in a dry pit in order to be easily accessible. The bearings of vertical pumps require considerable attention, and hence the horizontal-shaft type is used wherever feasible

244. Rotary Pumps. In their many forms, these mechanisms are distinguished from centrifugal pumps in that they are positive-displacement units which have close clearances not only between their moving parts but also between the internal assemblage and the casing which surrounds it. Rotary pumps have no valves, and barring undue leakages they can deliver a reasonably constant volume of liquid against pressures ranging from low to high. Their greatest field of application

is in pumping fuel oils and other liquids which have viscosity enough to prevent leakage and also some properties of lubrication.

Figure 203(*a*) illustrates a longitudinal section of a roller pump. When the pump is in operation, the rollers held within recesses of the revolving rotor tend to move outward by action of centrifugal forces. As indicated in Fig. 203(*a*), the liquid handled is entrapped between the rollers and the bottom of the casing and is discharged under pressure at the outlet connection.

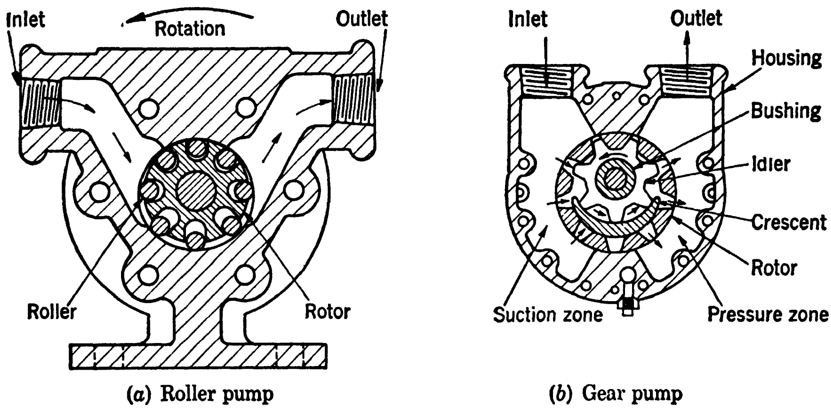


FIG. 203. Kraissl (*a*) roller and (*b*) gear pump.

The internal-gear type of pump of Fig. 203(*b*) has an overhung rotor which is forged integral with its shaft which is located at the center of the casing. The rotor has openings about its periphery which form nine internal teeth. Within the rotor is an idler gear with seven external teeth which mesh with those of the rotor. The idler gear is mounted upon a stud attached to the casing cover. The rotor and the idle gear revolve about different centers, so that the idler gear is eccentric with the rotor. The crescent-shaped piece at the bottom of the idler gear serves as a seal, and the liquid entrapped between the gear teeth is forced out against the discharge resistance.

Figure 204 is an illustration of the form of construction of a screw pump fitted with external drive gears. Such units are suitable for handling (with continuous flow) fuel and lubricating oils on shipboard and about power plants. The unit shown is capable of discharging from 1 to 700 gpm of viscous fluids against an outlet resistance of 1000 psig; when water is handled the discharge pressure at the rates of flow stated is 500 psig. The fluid impellers consist of left- and right-hand helical screw threads mounted upon a driving and a driven shaft.

The screw threads have close clearances between each other and also the pump housing. The threads are not in contact with each other. A part of the power applied to the driving shaft is transmitted to the idler shaft through cut herring-bone gears. Each end of either a driver or a follower shaft is held in its proper position by a suitable independent bearing.

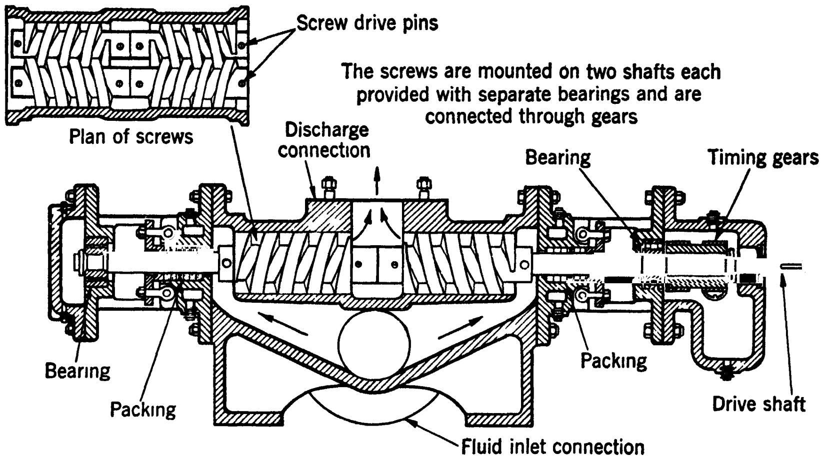


FIG. 204. Sier-Bath screw pump.

245. Performance of Pumps. The *capacity* of a pump is generally stated as the quantity of liquid discharged by it, expressed in gallons per minute. For a centrifugal pump the discharge varies directly as the speed. Reciprocating and plunger pumps may not discharge a volume of liquid equal to their piston or plunger displacements, in a given time. Such a condition of operation involves an item termed ***pump slip*** which is the difference between the displacement or theoretical discharge of a pump and the actual discharge. Pump slip is commonly expressed as a percentage of the displacement. Average values of pump slip range from 3 to 25 per cent. The *volumetric efficiency* of a piston pump is the ratio of the volume of fluid delivery to the pump displacement.

The *total head* against which a pump is delivering water is equal to the sum of the change of velocity head in the pump and the difference between the head with which the water leaves the pump and that with which the water enters the pump.

$$H_t = H_d - H_s + \frac{V_d^2 - V_s^2}{2g} \tag{235}$$

where H_t = total head, ft of water.
H_d = discharge head, ft of water.
H_s = suction head, ft of water.
V_d = velocity of water in discharge pipe, fps.
V_s = velocity of water in suction pipe, fps.
g = acceleration due to gravity, 32.17 ft per sec^2.

In equation 235 the small losses in the pump valves are neglected.

Suction lift is the vertical distance from the level of the water to be pumped to the center line of the pump. When the pump lifts the water from a level below the center line, the pressure in the suction line is less

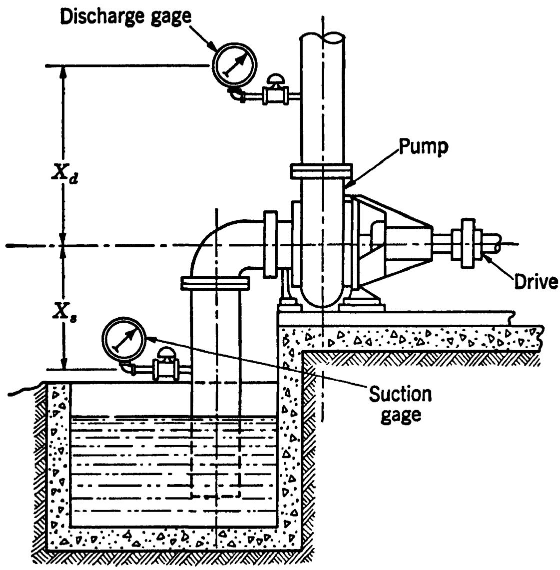


FIG. 205. Corrections for elevations of gages in a centrifugal-pump test.

than atmospheric, and H_s is considered negative. If the water level is above the center of the pump, the pump then receives water under pressure, and H_s is considered positive. Discharge head is the vertical distance between the center line of the pump and the level (or its equivalent) to which the water is elevated. The friction head may be determined by means of charts or tables, to be found in any standard handbook or textbook on hydraulics, which give the frictional losses in pipes, fittings, and valves. These frictional losses are necessarily approximate and depend on such factors as smoothness of pipe, fluid handled, length of time in service, type of valves, and radius of bends.

The velocity head, sometimes known as the head due to velocity, is the equivalent head through which the water would have to fall in order to acquire the velocity at a given point; that is, $H_v = V^2/2g$. The change of velocity head $(V_d^2 - V_s^2)/2g$ should be considered, however, when the discharge pipe is of different size (generally smaller) than the suction pipe, when the total head is low, and when the suction lift is high, or in accurate testing.

Figure 205 indicates possible locations of gages used for the determination of suction and discharge heads. The reading of the gage attached to the suction pipe gives an indication of the lift to the point of its attachment. In order to find the suction head H_s, the head equivalent of the suction-gage reading must have the distance X_s algebraically added to it. Likewise for the discharge head H_d when the discharge-head gage is above the center line of the pump, the distance X_d must be added to the feet of head represented by the gage reading, and when the gage is attached to a discharge pipe below the center line of the pump a deduction must be made for the distance, in feet, that the center of the gage is below the pump center line.

Example. The diameter of the discharge pipe of a boiler-feed pump is 4 in., and the suction pipe is 6 in. The water delivered is 250 gpm, and the temperature is 60 F. The discharge pressure near the pump is 190 psig, and the suction is 12 in. of mercury vacuum. Find the total head pump against, if the discharge gage is located 3 ft above the center of the pump and the vacuum-gage *connection* to the suction pipe is 2 ft below the center of the pump. Assume that the head necessary to overcome the frictional resistance through the pump is 2.20 ft.

Solution. Weight of water at 60 F = 62.34 lb per cu ft from Table XIII; see Appendix.

$$H_d = \frac{190 \times 144}{62.34} + 3 = 441.8 \text{ ft} \quad \text{discharge head}$$

$$H_s = \frac{-12 \times 13.59}{12} - 2 = -15.59 \text{ ft} \quad \text{suction head}$$

$$\text{Quantity flowing} = \frac{250 \times 231}{1728 \times 60} = 0.557 \text{ cfs}$$

From the equation,

$$Q = AV \text{ (quantity = area} \times \text{velocity)}$$

$$V_d = \frac{\text{cubic feet per second} \times 144}{\text{area of 4-in. pipe}} = \frac{0.557 \times 144}{12.73} = 6.30$$

$$V_s = \frac{\text{cubic feet per second} \times 144}{\text{area of 6-in. pipe}} = \frac{0.557 \times 144}{28.891} = 2.77$$

The pipe areas are from Table XIX of the Appendix.

$$H_{vd} - H_{vs} = (V_d^2 - V_s^2) \div 2g = (6.30^2 - 2.77^2) \div (2 \times 32.17) = 0.51 \text{ ft}$$

$$H_t = 441.8 - (-15.59) + 0.51 + 2.20 = 460.1 \text{ ft}$$

When an expression of total head p_t, psig, is desired the following procedure applies to the prior and other cases.

$$p_d = 190 + (3 \times 62.34)/144 = 190 + 1.3 = 191.3 \text{ psig}$$
$$p_s = (-12 \times 0.491) - (2 \times 62.34)/144 = -5.89 - 0.87 = -6.76 \text{ psig}$$
$$p_{vd} - p_{vs} = (0.51 \times 62.34)/144 = 0.22 \text{ psig}$$
$$p_f = (2.20 \times 62.34)/144 = 0.95 \text{ psig}$$
$$p_t = 191.3 - (-6.76) + 0.22 + 0.95 = 199.23 \text{ psig}$$

The hydraulic or *water horsepower*, whp, of a pump is the power actually delivered to the water as computed by either of two methods.

$$\text{whp} = \frac{m_w \times H_t}{33{,}000} \tag{236}$$

where m_w = water delivered by the pump, lb per min.
H_t = total head, ft of water.

Water horsepower can be calculated when the total head p_t, psig, and the volume of water handled Q, cfm, are used. The expression is

$$\text{whp} = \frac{144 \times p_t \times Q \times D}{33{,}000 \times D} = \frac{144 \times p_t \times Q}{33{,}000} \tag{237}$$

The *mechanical efficiency* of a pump is the ratio of the energy converted into useful work to the energy supplied to the pump; that is, the ratio of the water horsepower output to the shaft horsepower input to the pump, or

$$e_m = \frac{\text{whp}}{\text{shp}} 100 \tag{238}$$

The shaft (or brake) horsepower is considered as the power supplied to the pump shaft by motor, turbine, or pulley.

The *thermal efficiency* of a steam-driven pump is the ratio of the heat equivalent of the water horsepower to the heat supplied at the steam end. Another expression of the performance of a steam-driven pump is *duty* which may be stated as either the foot-pounds of work done per million Btu chargeable to the steam end or the foot-pounds of work done per thousand pounds of dry steam supplied to the pumping unit. Duty calculations are now of minor importance. The expression of

thermal efficiency, e_t in per cent, is stated as

$$e_t = \frac{2545 \times \text{whp} \times 100}{m_s(h_1 - h_{f,2})} \tag{239}$$

where m_s = weight of steam supplied, lb per hr.
h_1 = total enthalpy of steam supplied, Btu per lb.
$h_{f,2}$ = enthalpy of the liquid at exhaust pressure, Btu per lb.

Example. A 10 in. by 6 in. by 10 in. direct-acting duplex pump with 1½-in. piston rods makes 35 double strokes per cylinder per minute and delivers 135 gpm of 100 F water against a total head of 460 ft. The steam is supplied to the steam end at a pressure of 150 psig and exhausts to the atmosphere. Amount of steam supplied is 3800 lb per hr at 0.98 quality; barometer, 29.6 in. of mercury. Assume that the piston makes full stroke. Find (*a*) the piston displacement of the pump in cubic feet per minute, (*b*) pump slip, (*c*) water horsepower delivered, and (*d*) the thermal efficiency of the pump.

Solutior

(*a*) $$\text{Piston displacement} = \frac{0.7854[6^2 + (6^2 - 1.5^2)]}{144} \times \frac{10}{12} \times 35 \times 2 = 22.15 \text{ cfm}$$

(*b*) $$\text{Volumetric efficiency} = \frac{135 \times 231}{1728} \times \frac{1}{22.15} \times 100 = 81.5 \text{ per cent}$$

$$\text{Pump slip} = 100 - 81.5 = 18.5 \text{ per cent}$$

(*c*) $$\text{Water horsepower} = \frac{460 \times 135 \times 231 \times 61.99}{1728 \times 33{,}000} = 15.6$$

(*d*) $$\text{Thermal efficiency} = \frac{2545 \times 15.6 \times 100}{3800[338.3 + (0.98 \times 857.3) - 179.4]}$$

$$= 1.045 \text{ per cent}$$

The characteristic performance curves, see Fig. 206, for pumps are usually based upon a constant speed of the pump, the curves showing the relation between (*a*) head and capacity, (*b*) shaft horsepower and capacity, and (*c*) mechanical efficiency and capacity. The values used in plotting the above curves are those computed for a given definite speed of the pump, usually that at which the pump operates at the best efficiency or that at which it was designed to operate. For centrifugal pumps, the relation between the various quantities is largely controlled by the angles and the curvatures of the impeller blades and the shape of the volute or the arrangement and the design of the diffusion vanes. If the vanes are radial or inclined forward in the direction of rotation, the head will increase with increased capacity; if the vanes are curved backwards sufficiently, the head will remain constant or decrease as the capacity increases, as shown by Fig. 206. Centrifugal-pump and fan

characteristic curves are similar. In general, for a given centrifugal pump, the *quantity of water delivered varies directly with the speed*, the *head with the square of the speed*, and the *power with the cube of the speed.* Thus doubling speed of a pump impeller doubles the quantity of water pumped, produces a head four times as great, and requires eight times as much power to drive the pump.

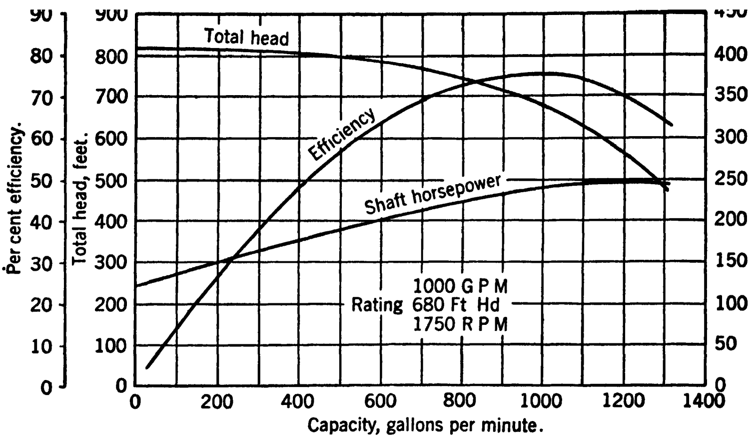


FIG. 206. Characteristic performance curves of a 6 in. by 5 in. (17-in. diameter impellers) three-stage centrifugal pump used for boiler feedwater.

Since manufacturers furnish curves for their specific output, and as pump-performance curves vary within wide limits, general curves are without purpose except for general approximations. In Fig. 206 the highest point on the total head curve is 820 ft; hence the pump could not be used for a greater head because at the existing constant speed no water would be delivered. The pump under consideration delivered 1000 gpm against a total head of 680 ft when operating at a speed of 1750 rpm. Under the conditions mentioned, 238 shp were necessary to drive the pump, and the mechanical efficiency was 75 per cent.

246. Usages of Piston and Centrifugal Pumps. *Piston pumps* have wide fields of application in small power plants and for certain industrial purposes, where the quantity of fluid to be handled is small and the total pressure pumped against is high, as with hydraulic presses. Where either the head or the discharge (quantity) is required to vary within wide limits and where they do not maintain definite relations to each other, the displacement pump will be found to be flexible and economical. Simple piston pumps are well adapted for vacuum pumps in connection with condensers because the steam-cylinder clearances are

comparatively small and there is little or no short-stroking. The practically continuous action of duplex pumps makes them better for high-pressure service than simple pumps; duplex pumps are also preferable where either a very high or a very low velocity of flow is required. Piston and plunger pumps cannot satisfactorily handle dirty, gritty, and highly viscous fluids.

The *centrifugal pump* with steam-turbine or electric-motor drive combines maximum efficiency in power transmission with simplicity, compactness, balanced rotary motion, absence of valves and pistons, uniform discharge, reliability, and small weight. For pumping small volumes of water against high pressures, the centrifugal pump can be connected directly to, and run at the same speed as, the turbine. Where the proper speeds of the pump and turbine are too far apart to permit satisfactory efficiency at a compromise speed, or where the quantity of water to be handled is very large, speed-reducing gears are interposed between the turbine and the pump. Boiler-freed pumps, condensate pumps, hydraulic-pressure pumps, elevator pumps, fire pumps, mine pumps, and other similar-purpose pumps are usually directly connected to the turbine. Condenser-circulating pumps, large general water-service pumps, and water-works pumps are generally driven by gears. The centrifugal pump discharges a continuous stream of fluid. Originally developed as a low-lift pump, the centrifugal pump is now used for high-pressure service, and pressures of 3500 psig can be obtained in a multistage unit. This type of pump is less costly to install, operate, and maintain than the piston type and will pump water containing mud and small stones. Some of the disadvantages of centrifugal pumps are that the efficiency is not usually as high as that of the best grade of piston pump and that the pump is not satisfactory where a high suction lift is necessary or for pumping small quantities of water against high discharge pressures. A centrifugal pump when designed for a certain capacity and total pressure cannot be used without considerable loss of efficiency at any other capacity or pressure. The reciprocating pump is more flexible in this respect, as it can be operated under widely different conditions without any large sacrifice of efficiency.

247. Steam Injectors. These devices with a single moving part (overflow valve) are able to discharge water against a pressure of 40 to 60 psig greater than the steam supplied to it. The internal construction of an injector and an example of its use in feeding water to a boiler are shown in Fig. 207. Steam entering the device acquires a high velocity in a nozzle, at the expense of some of its pressure, and imparts momentum to water flowing from the supply pipe. This action causes water to flow through the suction nozzle, the delivery nozzle, and the

discharge connection. In the lifting type of injector, shown in Fig. 207, a partial vacuum is produced as steam enters the suction nozzle where it condenses. As the steam loses velocity by impact the kinetic energy of the water is increased. Later with decreased velocity of water flow the kinetic energy is changed to potential energy, and a pressure is built up in the delivery tube or nozzle. The nonlifting type of injector must have the water supply above its inlet, as the device cannot create a

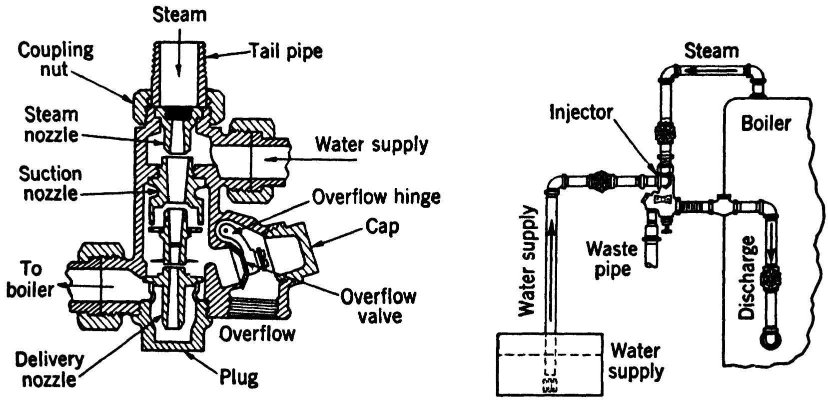


FIG. 207. Penberthy injector and its application to a steam boiler.

partial vacuum so that water will be forced upward in the suction pipe by the pressure of the atmosphere.

The efficiency of the injector from a thermodynamic standpoint is nearly 100 per cent, as practically all the heat of the steam (less a small radiation loss) is returned to the boiler. As a pump the efficiency of the injector is only about 2 per cent; its steam consumption is high, being around 400 lb per whp-hr, whereas a small steam-driven pump may require from 150 to 200 lb per whp-hr.

Inasmuch as an injector functions as a direct-contact feedwater heater, an energy balance may be written for it in the form of equation 185 of Art. 132 when radiation losses are ignored.

248. Ejectors. These appliances find a limited use for handling liquids where the suction lift is small and the resistance to discharge is not great. Figure 208 illustrates a water-jet eductor (ejector) which is capable of handling without clogging slurries or liquids containing semisolids. In principle the air and vapor ejectors described in Art. 229 are similar to the device of Fig. 208. This type of pumping mechanism is self-priming, has no moving parts, is capable of installation in restricted spaces, and is low in first cost.

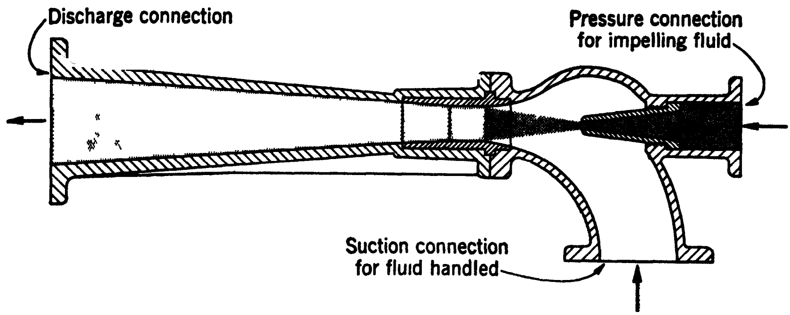


FIG. 208. Schutte and Koerting eductor.

249. Measurement of Water. Methods of water measurement are divided into primary and secondary, according to whether the method is an absolute measurement of quantity or merely the measurement of an effect of quantity or rate of flow. Of these methods the only primary methods are those of measurement by volume or weight. Secondary methods are more generally used in practice, and include measurement by Venturi tube, weir, orifice,[1] nozzle,[1] or pitot tube.[1]

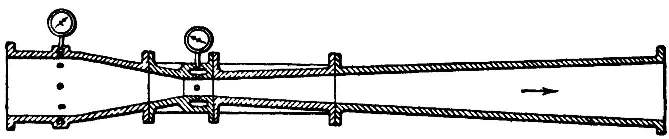


FIG. 209. Section through a Venturi tube.

For the measurement of flow in pipes under pressure the *Venturi tube*, Fig. 209, is a reliable form of meter and is extensively used where accurate and consistent results are desired. The tube takes the shape of two truncated cones, joined at their smallest diameters by a short throat section and inserted between flanges in a pipeline. Suitable pressure connections are provided for observing the pressure at the entrance and at the throat of the Venturi tube. The entrance cone has a total angle of about 21 deg and leads to the short cylindrical throat section which is accurately machined. The diameter of the throat is usually between one fourth and one half of the pipeline diameter. The exit end of the throat leads to the exit cone or diffuser which has a total angle of 6 deg. The maximum discharge through a Venturi tube is obtained when the throat pressure is 53 per cent of the entrance pressure.

[1] For additional information concerning these methods refer to "Fluid Meters, Their Theory and Application," *ASME Research Publications*.

The equation for determining the quantity of water flowing through a Venturi tube is

$$Q = \frac{CA_1 \sqrt{2gH_w}}{\left[\left(\frac{D_1}{D_2}\right)^4 - 1\right]^{1/2}} \qquad (240)$$

where Q = rate of discharge, cu ft per sec.
C = coefficient, usually 0.96 to 0.99.
A_1 = area of pipe at upstream pressure connection, sq ft.
g = acceleration due to gravity, 32.17 ft per sec.^{2}
H_w = difference in head, corresponding to the difference between the static pressures at the two "piezometer" connections, ft of water.
D_1 = diameter of pipe, upstream.
D_2 = diameter of pipe, throat.

The coefficient C varies with the rate of flow, diameter of the pipe, smoothness of inside surface, viscosity of the fluid, and other factors. In practice, the numerical value of C is determined from experimental tests.

Example. The inlet diameter of a Venturi tube is 6 in., and the throat diameter is 2 in. Water flowing through the tube has a temperature of 150 F, an inlet static pressure of 160 psig, and a throat static pressure of 96 psig. How many gallons per minute is the pump delivering? Assume coefficient as 0.99.

Solution.

$$H_W = \frac{(160 - 96)144}{61.2} = 150.6 \text{ ft of water}$$

From equation 240,

$$Q = \frac{0.99 \times 0.1963\sqrt{2 \times 32.17 \times 150.6}}{[(\frac{6}{2})^4 - 1]^{1/2}}$$

$$Q = \frac{19.13}{8.95} = 2.14 \text{ cfs}$$

or

$$= \frac{2.14 \times 60 \times 1728}{231} = 961 \text{ gpm}$$

The ***weir*** is an accurate means of determining the rate of flow of large quantities of water in open conduits. There are four principal types: the square notch; the rectangular weir; the V-notch, which increases the head readings at low rates of flow; and the parabolic notches, which are so shaped as to give head readings directly proportional to the flow. The rectangular weir is generally used for high heads, such as may be found in the measurement of large streams. When the heads

are relatively low (less than 12 in.), as in most power-plant service, the V-notch of Fig. 210 is more reliable. In the weir the variable area of the stream bears a definite relation to the velocity and is indicated by the height of the level of the measured liquid over a horizontal plane taken as a base. This variable height is known as the head, H_w, of the weir and is the difference of elevation in feet between the surface of the liquid in the channel at a point just ahead of the beginning of the sur-

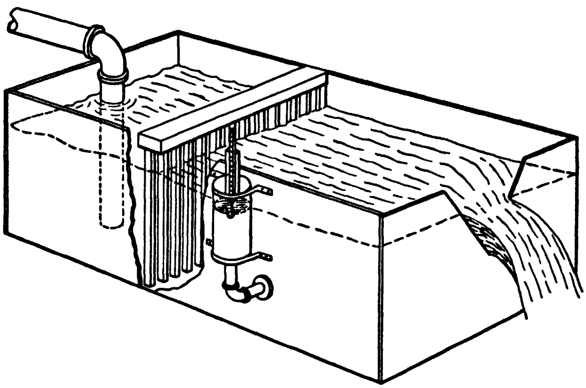


FIG. 210. V-notch weir with combing rack and hook gage.

face curve and the bottom or sill of the notch. The head of water passing over the V-notch is measured by a hook gage which is located at a point where flow occurs without turbulence and ahead of the beginning of the surface curve. The turbulence of the water flowing through a weir tank may be minimized by properly locating suitable baffles; the combing rack serves as an efficient means of straightening the flow.

For a triangular weir or V-notch, apex down, with angle at the apex of 90 deg, the rate of flow is

$$Q = 2.53 H_w^{5/2} \tag{241}$$

where Q = quantity of flow, cfs.
H_w = head of water passing over the weir, ft.

PROBLEMS

1. A small pump is operated under the following conditions: discharge-pressure gage, 3 ft above the pump center line; point of connection of the suction-lift manometer, 2 ft below the pump center line; reading of the discharge-pressure gage, 100 psi; reading of the suction manometer, 8 in. of mercury; density of water handled, 61.5 lb per cu ft; and velocity-pressure equivalent in the suction pipe, 0.75 psig less than that in the discharge pipe. Find the total head created by the

pump, in feet of fluid handled, when the friction losses in the piping between gage locations are neglected. 249.9 ft., *Ans.*

2. A 7 in. by 4½ in. by 10 in. duplex piston pump, with a 4-in. suction and a 3-in. discharge pipe is rated to handle 82 gpm of water at 60 F, when the pump operates with 40 single strokes per minute in each water cylinder. The discharge-pressure gage reads 60 psig, and the suction-lift manometer 20 in. of water when the pump is in normal operation. The suction manometer is attached to the pipe at a point 1.5 ft below the pump center line. A correction for gage elevation of 2 ft must be added to the discharge-gage head equivalent. Find the total head, psig, when friction losses in the piping are neglected between the gage locations.

3. A manometer, attached to the 1-in. discharge pipe of a screw pump, indicates an 18-in. difference of mercury-column elevations when its connection is 24 in. above and the lower mercury level is 6 in. below the centerline of the unit. Oil from the pump fills the manometer connection above the lower level of the mercury. A suction gage, 1 ft above the pump center line, shows a positive pressure of 1 psig in the 1¼-in. pipe. The pump handles 25 gpm of oil at a temperature of 80 F and a specific gravity of 0.88. The friction losses between gages represent 1 per cent of the calculated head. Find the total head against which the pump operates expressed in feet of fluid. 20.5 ft, *Ans.*

4. The liquid cylinder of a simplex pump has the following dimensions: diameter, 6 in.; stroke, 12 in.; and diameter of piston rod, 1¾ in. When the pump is operated with 40 double strokes per minute its volumetric efficiency is 92 per cent, when handling water. Find the volume of water discharged as measured in gallons per minute.

5. A 12 in. by 7¼ in. by 12 in. duplex piston pump with 2-in. piston rods makes 60 single strokes per cylinder per minute. The pump delivers 225 gpm of water when the average length of the strokes is 11½ in. Find the volumetric efficiency and the slip of the pump.

6. A pump discharges 250 gpm of water at a temperature of 180 F against a total head of 152 psig. Calculate the water horsepower developed by the unit.

7. The total head against which a steam-driven pump operates is expressed as 160 psig. The pump develops 84 ihp, discharges 720 gpm of water at a temperature of 180 F, receives steam of 0.97 quality at 140 psia, exhausts the steam against a pressure of 16 psia, and uses 145 lb of steam per hr per ihp. Find the water horsepower developed and the thermal efficiency of the unit as operated.

67.2 whp and 1.43 per cent, *Ans.*

8. A duplex direct-acting boiler feed pump is operated to give the following data: power developed in steam cylinders, 100 ihp; dry steam consumption per indicated horsepower hour, 120 lb; initial steam pressure, 150 psia; initial quality of steam, 0.98; exhaust pressure, 14 psia; water delivered, 805 gpm; water temperature, 915 F; and total head pumped against, 160 psig. Calculate the water horsepower and the over-all thermal efficiency based on brake horsepower.

75.2 whp and 1.57 per cent, *Ans.*

9. Calculate the water horsepower and the over-all thermal efficiency of a steam-driven pump which operates under the following conditions: water discharge, 600 gpm; water temperature, 160 F; water discharge pressure, 250 psig; power developed within steam cylinder, 100 ihp; steam quality, 0.98; steam pressure, 160 psia; exhaust pressure, 15 psia; and steam consumption, 120 lb per ihp-hr when wet. 87.2 whp and 1.86 per cent, *Ans.*

10. Find the hydraulic horsepower and the mechanical efficiency of a centrifugal

pump, direct-connected to a 5-hp electrical motor operating at full load, for the following conditions: fluid handled, oil of 0.85 specific gravity; oil temperature, 70 F; volume of oil handled, 300 gpm; and total head created by pump, 25 psig.

11. When operated at a speed of 1200 rpm a centrifugal pump gives the following performance: discharge of water, 400 gpm; and discharge head, 200 ft. If the pump is operated to give a discharge head of 500 ft without change of the system characteristic, what must be the speed and volume of water discharge?

12. A motor-driven pump is operated to give the following data: discharge pressure, 220 psig; water temperature, 100 F; water discharged, 340,000 gph; elevation of the discharge gage above the suction gage attachment, 36 in.; reading of the suction gage, 3 psi; diameters of the suction and the discharge pipes, equal; and motor input, 700 kw. Calculate the over-all efficiency of the unit.

13. (*a*) A pump handles brine having a temperature of 32 F and a specific gravity of 1.20 when the discharge pressure is 25 psig and the suction gage indicates a vacuum of 12 in. of mercury. The discharge gage is 24 in. above the pump center line, and the point of attachment of the suction gage is 10 in. below the pump center line. Find the total head of the pump in feet of fluid handled. (*b*) The mechanical efficiency of the motor-driven pump is 88 per cent. Find the brake-horsepower load of the motor when the pump handles 500 gpm of brine.

14. Steam of 0.96 quality is supplied to an injector at 170 psia, and is mixed with 12 lb of water per lb of steam. If radiation and friction losses are neglected, what final water temperature may be expected when the initial water temperature is 80 F?

15. A steam injector delivers 12 lb of water (including the weight of the steam) for each pound of steam supplied to it at 190 psia and 0.99 quality. The initial temperature of the water was 50 F. Calculate the heat required to deliver 1 lb of water to the discharge line. 90 Btu, *Ans.*

16. A pipeline is fitted with a Venturi meter which has an inlet diameter of 3 in. and a throat diameter of 1½ in. The unit operates with a static pressure difference of 5 psi between the inlet and the throat sections. (*a*) Find the discharge Q, in cubic feet per second of water at 70 F, when $C = 0.97$. (*b*) Find the necessary head on a 90-deg V-notch weir discharging the same quantity of water.

CHAPTER

14

Compression of Air and Gases

250. Uses of Compressed Air. Compressed air has a wide range of applications because of the convenience and ease of transmitting air as compared with steam. An important application is the operation of drills, hammers, hoists, sand blasters, controls, sprays, and pumps. Air compression is a major factor in the performance of internal-combustion engines and gas turbines. Compressors which differ only slightly from those for air are used in the transmission of natural gas and other gases by pipelines. The production of oxygen, nitrogen, and rare gases is accomplished by compressing and cooling to the liquefaction point.

The approximate air pressure required for pneumatic controls is 1 to 15 psig; for pneumatic tools, 70 to 90 psig; for air motors, 40 to 100 psig if the air is expanded; and for liquefaction of air as a means of separating its component gases, 2000 to 3500 psig.

251. Compressor Classification. For delivering air at low pressures, less than 15 in. of water, fans are generally used (see Chapter 9). Above this level various types of compressors and blowers are employed. These can be broadly classified as:

I. Positive displacement.
   1. Reciprocating compressors.
   2. Compression blowers.
   3. Noncompression blowers.

II. Nonpositive, or dynamic.
   4. Radial-flow centrifugal blowers.
   5. Axial-flow compressors.
   6. Mixed-flow compressors.

Any of these may be compounded in series or in parallel for the purpose of increasing pressure or capacity. This may be done by building dual, triple, or quadruple compound cylinders, or blowers, in one frame or arranging them as separate units with intercoolers.

Each type has particular features that enable it to excel in certain

applications. Thus, speed characteristic, capacity-weight ratio, or efficiency may mark a compressor as suitable for one use and not for another.

Reciprocating compressors require inlet and outlet valves and a piston with provision for lubricating the rings and cylinder walls. The compressed air will, therefore, contain a certain amount of oil. Centrifugal compressors or blowers do not require valves or internal lubrication, since there is no metal-to-metal contact.

252. Air Compressor Terminology. *Free air* is that which exists under the conditions of temperature and pressure at the compressor intake.

Capacity is the quantity of free air actually delivered by a compressor. It is usually measured in cubic feet per minute.

Piston displacement is the volume (cubic feet) obtained as the product of the piston area (square feet) and the piston stroke in feet.

Displacement per minute is the product of the piston displacement and the revolutions per minute. For two- or three-stage compressors the displacement is based on the low-pressure cylinder only, since it determines the amount of air passing through the other cylinders. In centrifugal compressors and rotary blowers, some slippage occurs and the actual capacity is frequently obtained by test rather than by calculation of displacement.

253. Reciprocating Compressors. The characteristics of a reciprocating compressor are such that it finds the widest application of any type. It is adaptable to small single-acting units such as paint sprays and filling-station tire pumps, Fig. 211, and to large compound units, Fig. 213, found in industrial, mining, and refrigeration installations. The average piston speed in the reciprocating compressor is limited, because of excess friction and lubrication problems, to approximately 1000 fpm in large compressors. Reciprocating compressors are satisfactory for all ranges of pressure. Single-stage units are used for pressures up to 150 psig, and three-stage units for pressures up to 3000 psig. Figure 212 shows a cutaway of a typical single-stage double-acting air compressor.

Figure 213 shows a double-acting two-stage compressor directly driven by a synchronous motor. This type of compressor is frequently found in an industrial plant where the synchronous motor helps to improve the power factor of the alternating-current electrical distribution system. The large rotating mass of motor usually eliminates the need for a fly wheel. The low starting torque of such motors requires that the compressor be started under no-load conditions; i.e., either the intake or the exhaust valves at each end of the cylinder are held open.

Synchronous motor-driven compressors are constant-speed machines; consequently, some means of unloading the compressor is necessary when the air is not being used. This may be accomplished by an

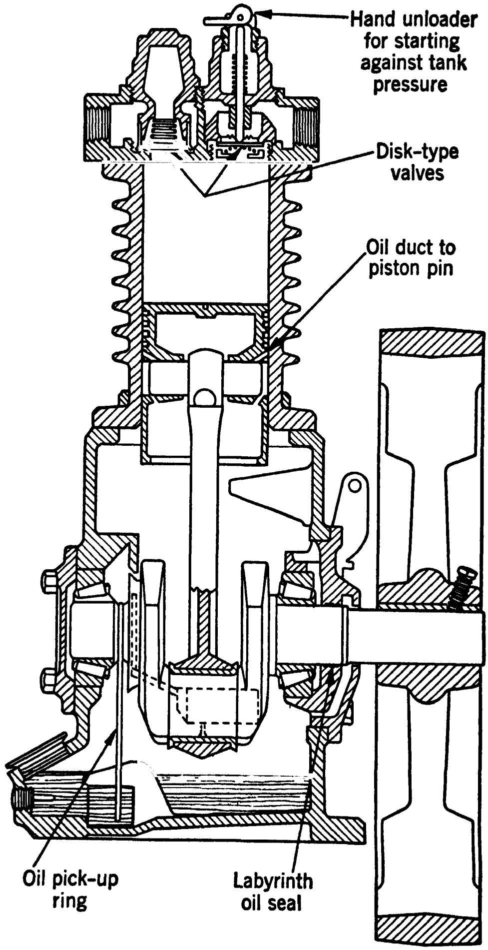


FIG. 211. Single-cylinder single-acting compressor.

automatic solenoid or hand-operated device which holds the intake valves open during the entire cycle, as in Fig. 214. One manufacturer regulates the constant-speed compressor by a five-step-clearance volume control, Fig. 215, which loads or unloads the compressor in successive steps. This variable-clearance control reduces the load as needed to meet the demand. Steam- and internal-combustion-engine-driven

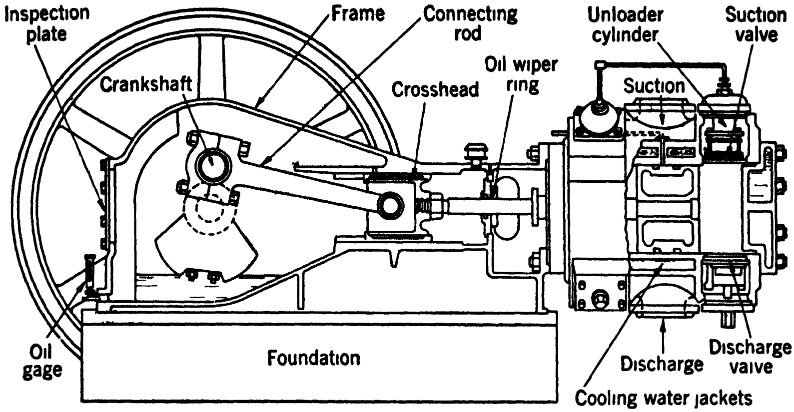


FIG. 212. Single-stage double acting compressor.



FIG. 213. Worthington double-acting two-stage compressor.

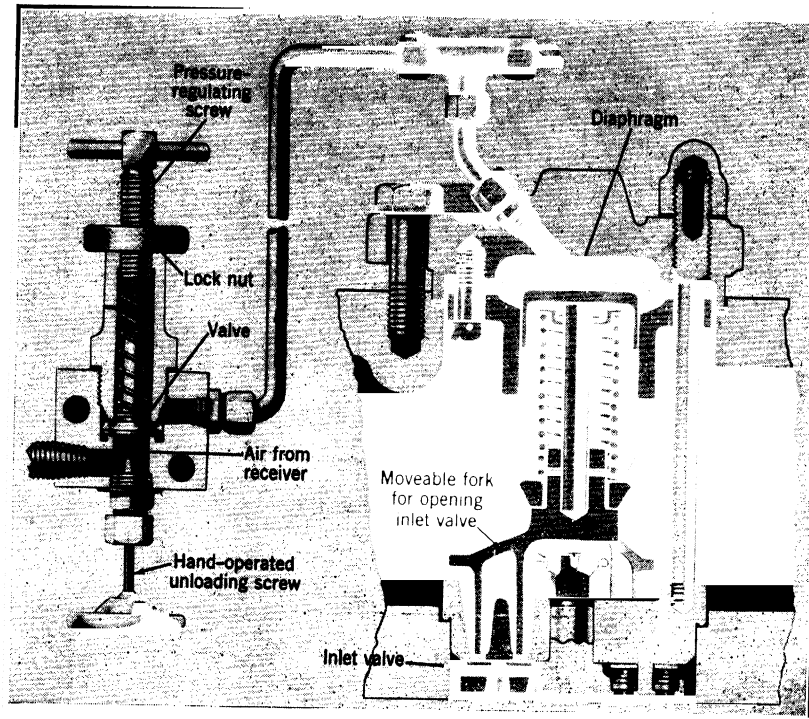


FIG. 214. Chicago Pneumatic inlet-valve unloading system.

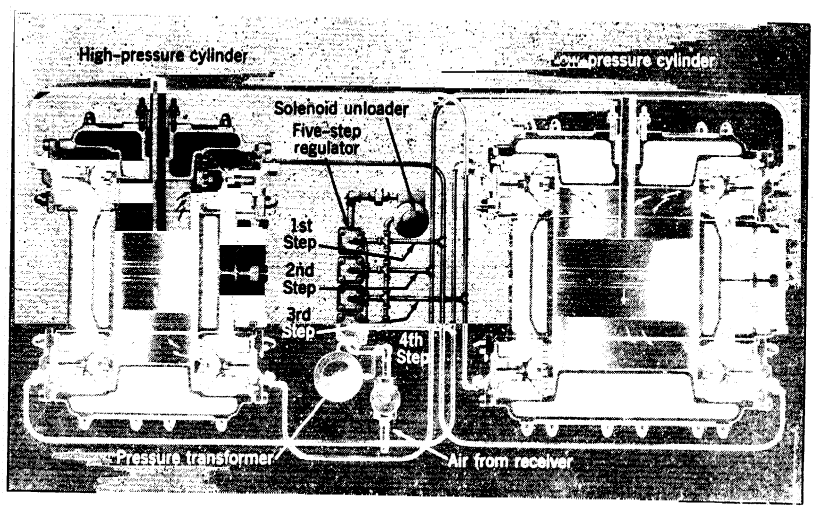


FIG. 215. Chicago Pneumatic five-step capacity regulator.

compressors are regulated by controlling the speed of the driving engine. Each end of each cylinder of double-acting compressors is fitted with inlet and discharge valves.

254. Ideal Reciprocating Compressors. The word "ideal" carries the usual assumptions, i.e., no friction losses, a perfect gas, and, in this case, no clearance volume. Such a compressor is shown in Fig. 216.

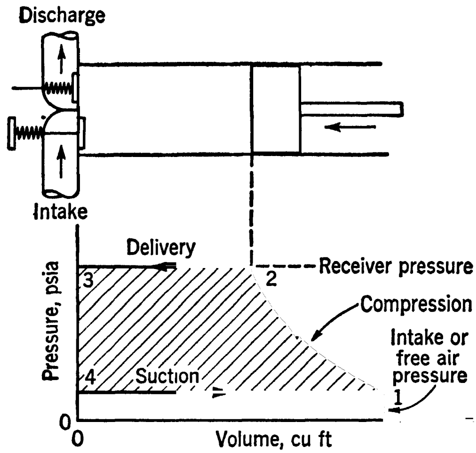


FIG. 216. Ideal air compressor with no clearance.

Area 1–2–3–4–1 is a picture of the work required to compress and deliver V_1 cu ft of free air; V_1 is also the displacement of this single-acting ideal compressor and is represented by the length of the line 4 to 1.

The work done during the cycle is the algebraic summation of the work around the cycle. Thus, from Fig. 216,

$$W \text{ per cycle} = {}_1W_2 + {}_2W_3 + {}_3W_4 + {}_4W_1$$

$$= \frac{(P_2V_2 - P_1V_1)}{1 - k} + (0 - P_2V_2) + (0) + (P_1V_1 - 0)$$

$$= \frac{(P_2V_2 - P_1V_1)}{1 - k} - (P_2V_2 - P_1V_1)$$

but $PV = mRT$. Therefore

$$W = \frac{k(P_2V_2 - P_1V_1)}{1 - k} = \frac{kmR}{1 - k}(T_2 - T_1) \tag{242}$$

where m = weight of air compressed, lb.

R = the individual gas constant, ft-lb$_f$ per (lb$_m$)(deg F).

It is more convenient to deal with intake and discharge pressures than with temperatures in air compressors of this type. For an isentropic process,

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{243}$$

Therefore, equation 242 may be written as

$$W = \frac{k}{1-k} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \text{ ft-lb} \tag{244}$$

where P_1 = absolute intake pressure, psfa.
P_2 = absolute discharge pressure, psfa.
$k = c_p/c_v = 1.4$ for air and diatomic gases.
$V_1 = mv_1$ = volume of free air delivered, or weight times specific volume at point 1, cu ft.
T = absolute temperature, deg F.

Equation 244 is the expression for the isentropic work of compressing V_1 cu ft of free air from P_1 to P_2. It is not limited to one cycle since V_1 equals mv_1, the product of *any* weight and *any* specific volume. When actual values are substituted in it, equation 244 will yield a negative quantity, thus indicating work done on the medium. Equation 244 may also be written as

$$W = \frac{k}{1-k} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \text{ ft-lb} \tag{245}$$

Equations 244 and 245 are the general expressions for the work of a single-stage compressor with isentropic compression. Since all conditions may be fixed, it is possible to compile tables of work or power required to compress any gas between given pressure limits. Such information may be found in handbooks under "isentropic horsepower." Equations 244 and 245 are applicable to units of various speeds and sizes. If it is assumed that the process is one of steady flow, then, per pound of medium,

$$\frac{P_1v_1}{J} + u_1 + \frac{V_1^2}{2gJ} + Q = \frac{P_2v_2}{J} + u_2 + \frac{V_2^2}{2gJ} + \frac{W}{J} \tag{246}$$

where all terms are as defined in Art. 30. *Note that in the steady-flow equation V represents velocity, whereas in other instances it may represent total volume.*

If the inlet velocity V_1 and the outlet velocity V_2 are the same and Q

is zero (adiabatic), then

$$\frac{W}{J} = \left(\frac{P_1v_1}{J} + u_1\right) - \left(\frac{P_2v_2}{J} + u_2\right)$$

or

$$\frac{W}{J} = h_1 - h_2 = c_p(T_1 - T_2) \quad \text{Btu per lb} \qquad (247)$$

Thus, in Fig. 216, area 1–2–3–4–1 pictures the change in enthalpy for the process. For unit weight flow, P_1v_1 is the entering flow work,

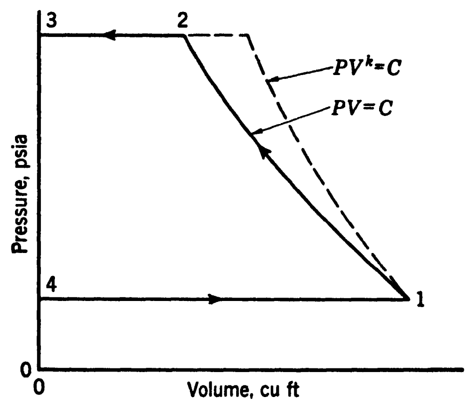


FIG. 217. Isothermal compression.

$(u_1 - u_2)$ is the change in internal energy during the compression, and $-P_2v_2$ is the flow work out. Since h_2 will be greater than h_1, W/J will be negative.

If the nature of the compression is such that $PV^n = C$, n being greater than 1, equation 244 becomes

$$W = \frac{n}{1-n} P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{(n-1)/n} - 1\right] \quad \text{ft-lb} \qquad (248)$$

where W is the work necessary to compress and force V_1 cu ft of free air out of the cylinder. Equation 248 is called the *polytropic equation* for the work of compression.

If the temperature during compression is held constant by cooling, n in the equation $PV^n = C$ has a value of unity or $PV = C$, which represents isothermal compression as shown in Fig. 217. Equation 248 becomes indeterminate when n is equal to unity. The net work for this case can be determined as before by a summation of the work around

the cycle. For a single-stage unit without clearance

$$W = {}_1W_2 + {}_2W_3 + {}_3W_4 + {}_4W_1$$

$$= \int_1^2 P\,dV + (0 - P_2V_2) + (0) + (P_1V_1 - 0)$$

but

$$PV = P_1V_1 = C$$

$$P = \frac{P_1V_1}{V} = \frac{C}{V}$$

Therefore

$$W = P_1V_1\int_1^2 \frac{dV}{V} - (P_2V_2 - P_1V_1)$$

or

$$W = P_1V_1 \log_e \frac{V_2}{V_1} = P_1V_1 \log_e \frac{P_1}{P_2} \tag{249}$$

In equation 249 reversibility is assumed but the compression path can be described as isothermal.

Figure 217 shows that isothermal compression is desirable since the work required is decreased. Isothermal compression can be approached in a slow-speed compressor having a clean water jacket and using an ample amount of cold water. Compression is normally too rapid to allow time for much heat transfer, so that in high-speed compressors the adiabatic is approached or exceeded.

255. Ideal Multistage Compressors. Compressors are built in multistage units so that intercoolers may be incorporated. Although water jackets may be used on all cylinders it is impossible to cool the gas adequately without passing it through an intercooler between stages. In addition to the usual assumptions of isentropic compression, the *ideal* multistage compressor assumes perfect control of pressure and heat transfer in the intercooler.

In Fig. 218 the ideal diagrams are shown for the low- and high-pressure cylinders of a single-acting two-stage compressor. The low-pressure diagram is shown as 1–2–3–4–1, and the high-pressure diagram is 1′–2′–3′–4′–1′. Air flows into the low-pressure cylinder from 4 to 1, and isentropic compression occurs from 1 to 2. The air is then forced to flow from the low-pressure cylinder at 3. The air next enters the intercooler where the temperature is reduced to its original value corresponding to point 1. On leaving the intercooler the air must be drawn into the high-pressure cylinder, 4′ to 1′, where its volume, pres-

sure, and temperature are represented by point 1′. Compression is then resumed from 1′ to 2′, after which the air is forced to flow into the receiver at point 3′. The reduction in work due to the intercooler is shown by the crosshatched area 1′–2′–5–2-1′ in Fig. 218(*a*). The energy that must be extracted in the intercooler is indicated as *a*–1′–2–*b*–*a* in Fig. 218(*b*).

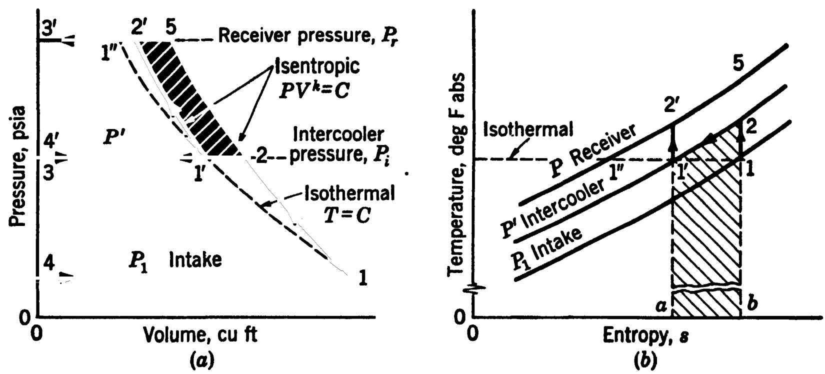


FIG. 218. *PV* and *Ts* diagrams of ideal two-stage compressor.

For a two-stage compressor the minimum work occurs when both cylinders do the same amount of work. Since the high-pressure cylinder must receive all the air delivered by the low-pressure cylinder, the intercooler pressure will be dictated by the relative sizes of the two cylinders. The total work is $W_{LP} + W_{HP}$. Equation 244 applies to either cylinder, and when $W_{LP} = W_{HP}$

$$\frac{k}{1-k} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] = \frac{k}{1-k} P'_1 V'_1 \left[\left(\frac{P'_2}{P'_1}\right)^{(k-1)/k} - 1\right] \tag{250}$$

and

$$\frac{P_2}{P_1} = \frac{P'_2}{P'_1}$$

Also $P_2 = P'_1 = P_i =$ *intercooler pressure.* Therefore,

$$P_i^2 = P_1 P'_2 \quad \text{or} \quad P_i = \sqrt{P_1 P'_2} \tag{251}$$

The best intercooler pressure P_i is thus the square root of the product of the intake and receiver pressures. The same result will be obtained by taking the first derivative of the total-work expression with respect to the intercooler pressure and equating it to zero.

When the work in each cylinder is the same, the total work is twice that of either cylinder, or

$$W = \frac{2k}{1-k} mRT_1 \left[\left(\frac{P_r}{P_1} \right)^{(k-1)/2k} - 1 \right] \tag{252}$$

where P_r = the *receiver pressure.*

Likewise in a three-stage compressor each cylinder should do one-third of the work, or

$$W = \frac{3k}{1-k} P_1 V_1 \left[\left(\frac{P_r}{P_1} \right)^{(k-1)/3k} - 1 \right] \tag{253}$$

Example. Find (*a*) the isentropic work per minute and (*b*) the isentropic horsepower required to compress 10 lb of air per min from 14.7 psia and 60 F to 100 psia in a single-stage compressor.

Solution. (*a*)

$$W \text{ per min} = \frac{k}{1-k} mRT_1 \left[\left(\frac{P_2}{P_1} \right)^{(k-1)/k} - 1 \right]$$

$$= \frac{1.4}{1-1.4} \times 10 \times 53.3 \times 520 \left[\left(\frac{100}{14.7} \right)^{(1.4-1)/1.4} - 1 \right]$$

$$= -707{,}000 \text{ ft-lb}$$

(*b*) $$\text{hp} = \frac{W \text{ per min}}{33{,}000} = \frac{-707{,}000}{33{,}000} = -21.4$$

The negative sign indicates work done on the air.

Example. Find (*a*) the isentropic work per minute and (*b*) the isentropic horsepower necessary to compress 10 lb of air per min from 14.7 psia and 60 F to 100 psia in an ideal two-stage compressor.

Solution. (*a*)

$$W \text{ per min} = \frac{2k}{1-k} mRT_1 \left[\left(\frac{P_r}{P_1} \right)^{(k-1)/2k} - 1 \right]$$

$$= \frac{2 \times 1.4}{1-1.4} \times 10 \times 53.3 \times 520 \left[\left(\frac{100}{14.7} \right)^{(1.4-1)/2.8} - 1 \right]$$

$$= -610{,}000 \text{ ft-lb}$$

(*b*) $$\text{hp} = \frac{W \text{ per min}}{33{,}000} = \frac{-610{,}000}{33{,}000} = -18.45$$

Note that the intercooler provides a reduction of (21.4 − 18.45) or 2.95 horsepower over the single-stage compressor used in the preceding example for the same conditions.

256. Actual Indicated Work. The actual indicator diagram departs considerably from the ideal, the real compressor requiring a

greater expenditure of work. The difference is due to intake and discharge pumping losses, turbulence, friction, leakage, heat transfer, and flow resistance in the intercooler.

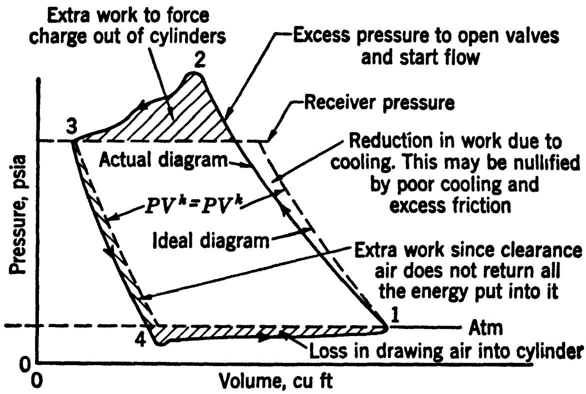


FIG. 219. Actual and ideal single-stage compressor diagram.

From Fig. 219 it becomes apparent that, in order to keep the work area to a minimum, the compressor should have large, easily opened intake and exhaust valves, good cooling, clean intake filter (low resistance to flow), and large intake and discharge piping. If the valves are to function quickly, they should be light weight with small inertia effects. Therefore most compressors of the reciprocating type have valves made of light steel strips or rings that are easily lifted from the valve seat by a small difference in pressure (see Fig. 220).



FIG. 220. Ingersoll-Rand channel valve.

In Fig. 221 the combined indicator diagram is shown in which some causes of the difference between ideal and actual work are indicated by crosshatching. A combined diagram is made by putting the diagrams on the same set of pressure-volume axes. In most cases the high-pressure diagram is transferred point by point to the same scale of pressure

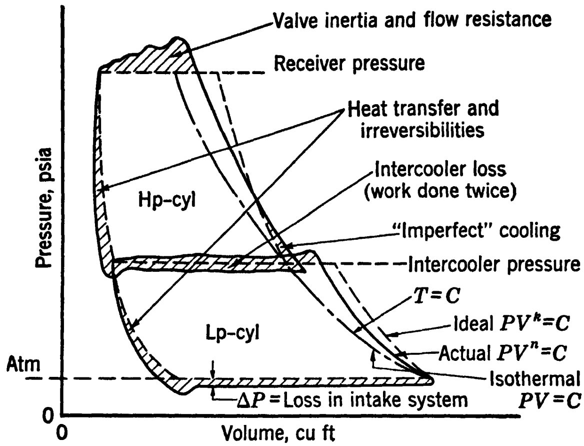


FIG. 221. Actual and ideal combined diagrams.

and volume as the low-pressure diagram. When taken from the compressor both diagrams will have approximately the same length, and are therefore to a different scale of volumes; also, the pressures will be measured by springs having different scale moduli.

257. Clearance Volume Effects. *Clearance volume* is the volume in the cylinder when the piston is at head-end dead center. *Per cent clearance* is the ratio (expressed as a percentage) of the clearance volume to the piston displacement. The clearance volume in reciprocating compressors, V_3 in Fig. 222, allows the compressed air to re-expand during the suction stroke to some point such as 4. The useful or effective stroke is thereby reduced to $V_1 - V_4$, Fig. 222, because no free air will enter the cylinder until the pressure has fallen to that of the intake.

If the exponent for the expansion curve 3 to 4, Fig. 222, has the same value as that for the compression curve 1 to 2 (k in the ideal case), the work required for compression with clearance is the same as that for compression without clearance and is given by equation 244.

Since reversibility was assumed in developing equation 244, the effect of clearance air is similar to that of a perfect spring which returns all the energy put into it. Theoretically, clearance volume does not change the amount of work required to compress a given amount of perfect gas.

Actually, heat transfer and irreversibility would prevent some of the energy put into the clearance air from being recovered.

In a double-acting compressor the clearance may not be the same at both head and crank ends. The clearance volume can be determined by

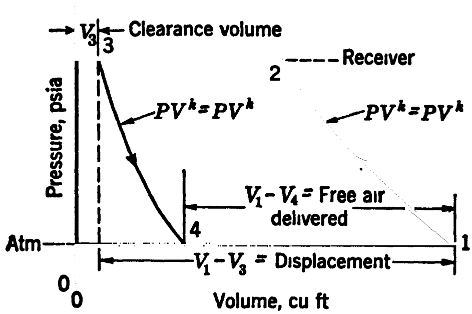


FIG 222 Ideal compressor with clearance.

filling the space with a measured amount of fluid when the piston is at dead-center position.

258. Positive Displacement Blowers. Compressors of the type shown in Figs. 223 (Roots) and 269 (General Motors) find a wide

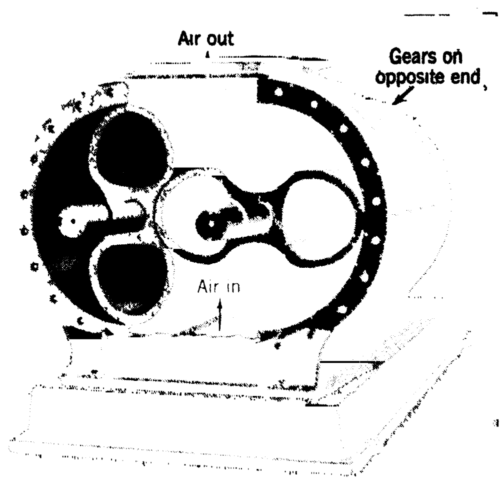


FIG 223 Roots-Connersville rotary positive blower, showing rotating impellers fitted to each other, and to cylinder, with clearances of a few thousandths of an inch.

application as scavenging pumps for two-stroke Diesel engines. When this type of compressor is directly geared to the engine, the air delivered

is proportional to speed, and thus, at low speeds (when a centrifugal unit would be at a disadvantage), this compressor will provide the air necessary for good torque characteristic, i e., maximum air to burn the maximum amount of fuel and thereby obtain maximum mean effective pressure and high torque.

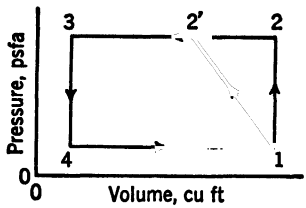


FIG 224 Work diagrams for positive displacement blower

In Figs 223 and 269 the rotors do not make actual contact with each other or with the casing, consequently, no internal lubrication is required and the air stream does not carry lubricating oil. Lubrication is the simple matter of supplying oil to a set of gears and four external bearings

Positive displacement blowers displace a fixed amount of air each revolution, and usually no compression occurs until the discharge port is uncovered and the backflow of air from the receiver causes compression. The indicator diagram would appear, therefore, as 1–2–3–4–1 in Fig. 224.

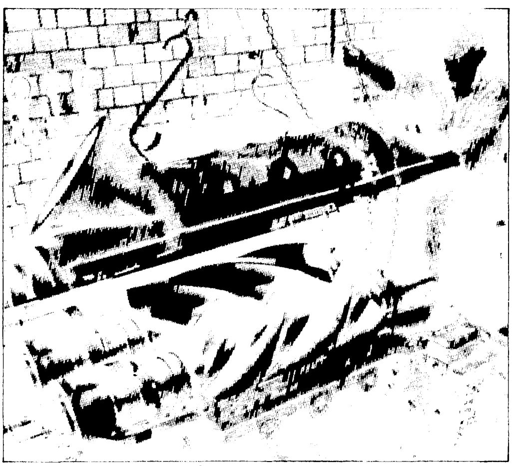


FIG 225 Elliott-Lysholm 12,500-cfm compressor

In the compressor shown in Fig. 225, air enters the bottom of the casing and is trapped by successive pairs of helical lobes As the rotors turn, the air is compressed until the discharge port is uncovered. The work diagram appears, therefore, as 1–2′–3–4–1 in Fig. 224 with a considerable saving in work over the noncompression type.

259. Centrifugal Compressors. Centrifugal compressors, Fig. 226, are high-speed machines (3000 to 40,000 rpm), and they are frequently

driven by turbines which have similar speed characteristics. Such compressors are often used as superchargers for internal-combustion engines and in many other applications where a large supply of air or gas is needed at relatively low pressures.

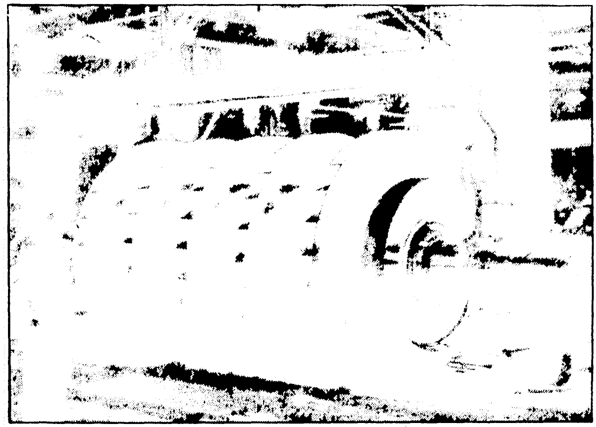


FIG. 226. Rotor of a Brown Boveri blast-furnace centrifugal blower.

A centrifugal compressor consists of a rotating impeller and one or more divergent passages through which the air is discharged. Air is

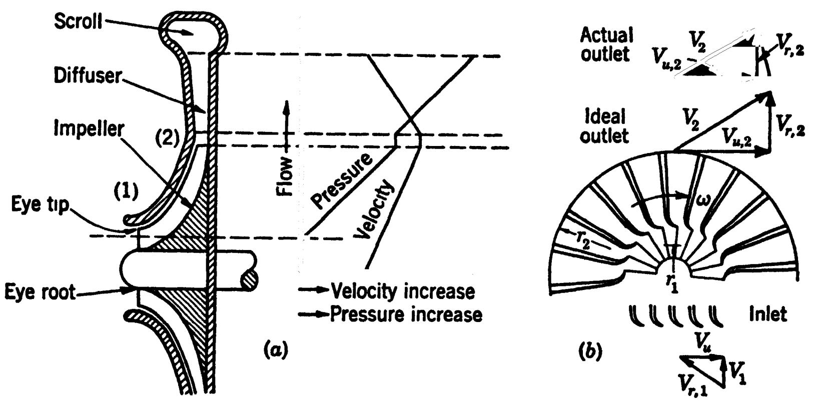


FIG. 227. Pressure and velocity relations in a centrifugal compressor. Subscript (1) denotes the inlet and (2) the outlet.

drawn in at the inlet or eye, Fig. 227(*a*), of the impeller and is given a whirling motion at high velocity by the impeller vanes and is flung out by centrifugal force. As the air passes through the impeller the centrip-

etal acceleration is obtained by a pressure head so that the static pressure increases from the eye to the tip of the impeller. As the air is flung off the impeller tip it is moving at high velocity and therefore has kinetic energy, most of which can be converted to pressure if the air is

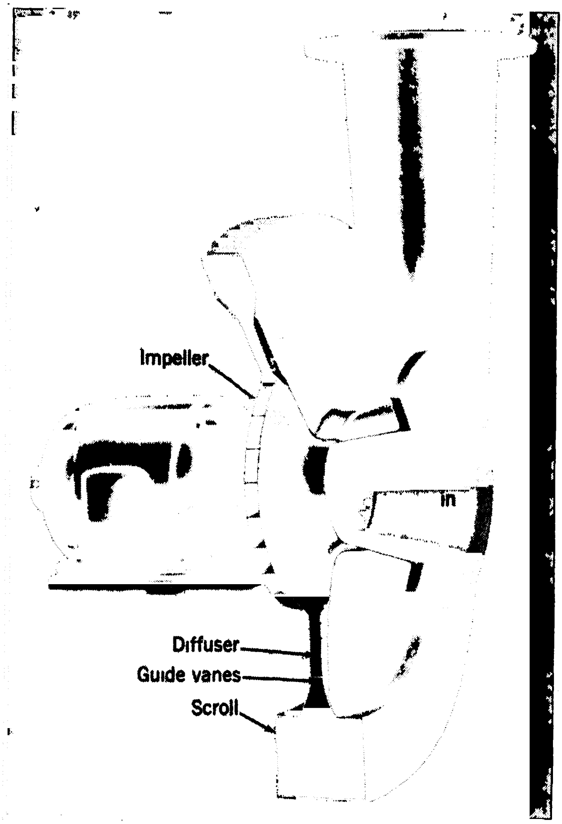


FIG. 228 Ingersoll-Rand single-stage centrifugal blower.

efficiently slowed down. It is the function of the divergent passage or *diffuser* to convert the kinetic energy to pressure. About half of the total pressure rise per stage is usually obtained in the impeller and half in the diffuser. It is much easier to obtain efficient acceleration than efficient deceleration of flowing gases, because the flow stream tends to break away from the passage walls with each change in direction or enlargement of the passage. Turbulence and eddy currents must be prevented because they convert the kinetic energy to heat rather than to pressure.

To prevent shock losses at the entrance to the impeller the leading edge of each blade is curved forward as shown in Fig. 227(*b*). The velocity of the impeller is $V_{u,1}$, and the absolute velocity of the air is given by V_1; the relative velocity between the impeller blade and the air is given by $V_{r,1}$. At point 2, for the ideal case, the air leaves the impeller with an ideal absolute velocity V_2 and a tangential or whirl velocity of $V_{u,2}$ equal to that of the impeller and with a radial velocity of $V_{r,2}$. It is assumed that no slip occurs; i.e., the air will be moving as fast as the rotor. Actually some slip does occur because inertia of the air causes it to resist radial acceleration and to "pile up" against the leading face of the blade. Unless the impeller is of the closed type, Fig. 228, some air will slip between the blades and the casing. The actual velocity diagram at the exit is influenced by the slip as indicated by Fig. 227(*b*). The slip reduces the actual exit velocities.

The theoretical torque (T) required to rotate the impeller per pound of air flow per second is

$$T = \frac{(V_{u,2})(r_2)}{g} \quad \text{ft-lb} \tag{254}$$

If the angular velocity in radians per second is ω, the theoretical work per pound of air is

$$W = \frac{(V_{u,2})(r_2)(\omega)}{g} \quad \text{ft-lb per sec} \tag{255}$$

or

$$W = \frac{(V_{u,2})(U)}{g} \quad \text{ft-lb per sec} \tag{256}$$

where $V_{u,2}$ = radial velocity of air leaving rotor, fps.
r_2 = radius of impeller tip, ft.
U = velocity of impeller tip, fps.

Or, if the slip is designated by $s = \dfrac{V_{u,2}}{U}$; $V_{u,2} = sU$, for each pound of flow

$$W = \frac{sU^2}{g} \quad \text{ft-lb per sec} \tag{257}$$

Additional losses such as windage and friction between rotor and casing require that additionl torque be supplied over that theoretically required. If the factor f, called the *power input factor*, is introduced to

account for this, the work equation becomes

$$W = \frac{fsU^2}{g} \quad \text{ft-lb per sec} \tag{258}$$

where f = power input factor, approximately 1.04 in modern designs.
s = slip factor, approximately 0.9 for open impellers.
U = velocity of impeller tip, fps.
W = work, ft-lb for 1 lb of flow per sec.

Example. A centrifugal compressor is to deliver 5 lb of air per sec. If the tip speed is 1500 fps, the slip factor 1.04, and the power input factor 0.9, find the horsepower delivered to the air.

Solution.

$$W = \frac{fsU^2}{g}$$

$$W = 5 \times 0.9 \times 1.04 \times \frac{1500^2}{32.2}$$

$$= 327{,}000 \text{ ft-lb per sec}$$

$$\text{hp} = \frac{327{,}000}{550} = 595$$

In using equation 258 in the above example, no allowance was made for bearing friction which is very small in a centrifugal machine. Neglecting this factor, the shaft power that would be delivered to the air stream is 595. Since the process is one of steady flow, this energy input must appear in some other form in the air. For unit weight flow per second between the compressor inlet point 1 and the discharge point 2, neglecting radiation, the steady-flow equation is

$$\frac{W}{J} = \left(u_1 + \frac{P_1v_1}{J} + \frac{V_1^2}{2gJ}\right) - \left(u_2 + \frac{P_2v_2}{J} + \frac{V_2^2}{2gJ}\right)$$

$$\frac{W}{J} = h_{T,1} - h_{T,2} \quad \text{Btu per lb per sec} \tag{259}$$

where $h_T = u + \dfrac{Pv}{J} + \dfrac{V^2}{2gJ}$ = *total* or *stagnation enthalpy*, Btu per lb.

Since, by equation 74 for a perfect gas,

$$h_1 - h_2 = c_p(T_1 - T_2)$$

it follows that

$$\frac{W}{J} = h_{T,1} - h_{T,2} = c_p(T_{T,1} - T_{T,2}) \quad \text{Btu per lb per sec} \tag{260}$$

where T_T = the *total* or *stagnation temperature*, deg F abs. These temperatures would be read on thermometers placed at the inlet point 1 and the outlet point 2 if the moving air which strikes them is brought to rest. Thermometers and thermocouples are commonly used in compressor work to determine the stagnation temperature rise. By applying equation 260 the horsepower and work input may be calculated per pound of air flow.

Example. The stagnation temperature at a compressor inlet is 520 deg F abs and at the compressor outlet is 856.5 deg F abs. Assuming the mean specific heat c_p to be 0.25 Btu per (lb) (deg F), find the horsepower required if 5 lb of air are compressed per sec.

Solution.

$$\frac{W}{J} = c_p(T_{T,1} - T_{T,2})$$

$$= 0.25(520 - 856.5) = -84.2 \text{ Btu per lb per sec}$$

$$\text{hp} = \frac{-84.2 \times 5 \times 778.26}{550} = -595$$

Again, if bearing losses are neglected, this is the shaft horsepower.

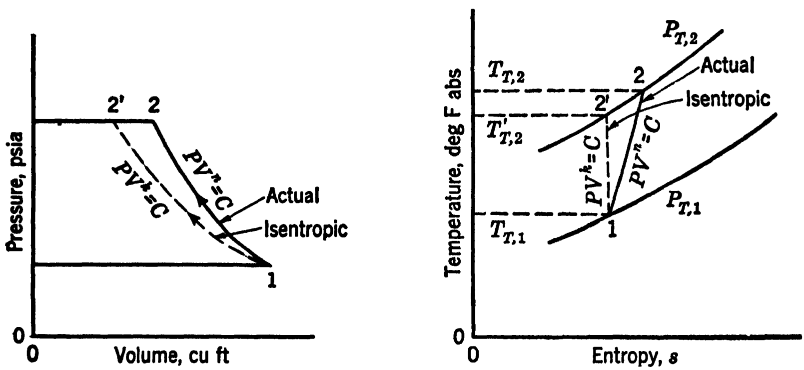


FIG. 229. Paths of compression for gases.

The efficiency of this horsepower input in producing a pressure rise in contrast to only a temperature rise[1] is dependent on the path of compression as illustrated in Fig. 229; and its measure is called *total-head isentropic efficiency* (e_i).

$$e_i = \frac{T'_{T,2} - T_{T,1}}{T_{T,2} - T_{T,1}} \tag{261}$$

[1] A temperature rise could be caused by heating only or by an impeller of zero efficiency with all the power input converted to friction and then to heat.

where $T'_{T,2}$ = stagnation temperature after isentropic compression, Fig. 229.

If compression is isentropic, then all the shaft input is delivered as work to the air stream and none as heat. The compression path would be $T_{T,1}$ to $T'_{T,2}$, Fig. 229, and the isentropic efficiency e_i would equal 100 per cent.

From equations 258 and 260 it follows that for the actual path

$$T_{T,2} - T_{T,1} = \frac{fsU^2}{gJc_p} \tag{262}$$

For the isentropic path

$$\frac{P_{T,2}}{P_{T,1}} = \left(\frac{T'_{T,2}}{T_{T,1}}\right)^{k/(k-1)} = \left[1 + e_i\left(\frac{T_{T,2} - T_{T,1}}{T_{T,1}}\right)\right]^{k/(k-1)} \tag{263}$$

where P_T = total or stagnation pressure, psfa.

T'_T = stagnation temperature after isentropic compression, deg F abs.

k = 1.4 = c_p/c_v for air.

e_i = isentropic efficiency, per cent expressed decimally.

s = slip factor, approximately 0.9.

f = power input factor, approximately 1.04.

c_p = mean specific heat, Btu per (lb) (deg F).

U = impeller tip velocity, fps.

The value e_i expresses the efficiency of compression as compared to the isentropic path and may reach 83 per cent for well-designed centrifugal compressors and 90 per cent for axial-flow compressors. From equations 262 and 263, it follows that

$$\frac{P_{T,2}}{P_{T,1}} = \left[1 + e_i\left(\frac{fsU^2}{gJc_pT_{T,1}}\right)\right]^{k/(k-1)} \tag{264}$$

Example. Find the pressure ratio and the stagnation discharge pressure that may be expected from a single-stage centrifugal compressor operating at a tip speed of 1400 fps if the isentropic efficiency = 83 per cent, the slip factor = 0.9, the power input factor = 1.04, the inlet stagnation temperature = 520 deg F abs, the mean specific heat = 0.24, and the stagnation pressure at inlet = 14.7 psia.

Solution. From equation 264

$$\frac{P_{T,2}}{P_{T,1}} = \left[1 + 0.83\left(\frac{0.9 \times 1.04 \times 1400^2}{32.174 \times 778.26 \times 0.24 \times 520}\right)\right]^{1.4/0.4}$$

$$\frac{P_{T,2}}{P_{T,1}} = (1.4875)^{3.5} = 4.0$$

$$P_{T,2} = P_{T,1} \times 4.0 = 14.7 \times 4.0 = 58.8 \text{ psia}$$

Single-stage compressors have their greatest efficiency when the delivery pressure ratio is approximately 2.5 to 1. For pressure ratios above 4 to 1, multistaging is employed as shown in Fig. 230. Such units are built with capacities as large as 100,000 cfm and to provide

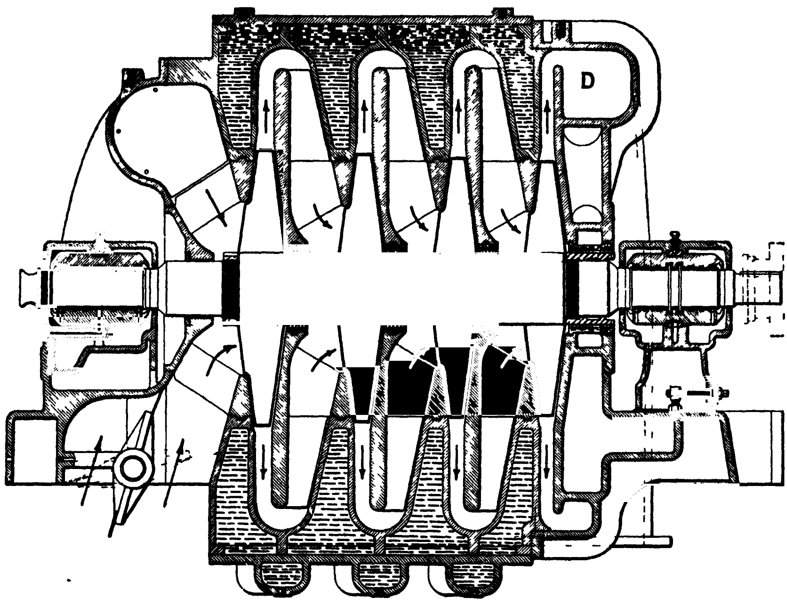


FIG. 230. Section of an Elliott four-stage centrifugal compressor.

pressures as high as 125 lb. One disadvantage of any high-speed machine is the resulting noise level which has limited the use of centrifugal compressors, although noise mufflers can be employed with considerable effectiveness.

Typical performance curves for a centrifugal compressor are shown in Fig. 231. A compressor can be designed for only one best point of operation called the *design point* for which it will have a definite capacity, speed, and delivery pressure. Although it may be operated at other than the design point, the resulting discharge pressure, capacity, and power required follow in general the fan laws at low pressures; i.e.,

1. The capacity varies directly as the speed ratio.
2. The total pressure varies as the square of the speed ratio.
3. The power input varies as the cube of the speed ratio.

Centrifugal compressors are not subject to control by throttling over a very wide range. Figure 231(*a*) shows that, if the speed is held con-

stant and the capacity is decreased by throttling, the operating point moves to the limit of stability or *surge line*. At the surge line the air stream is first moved forward by the impeller; then, because of the

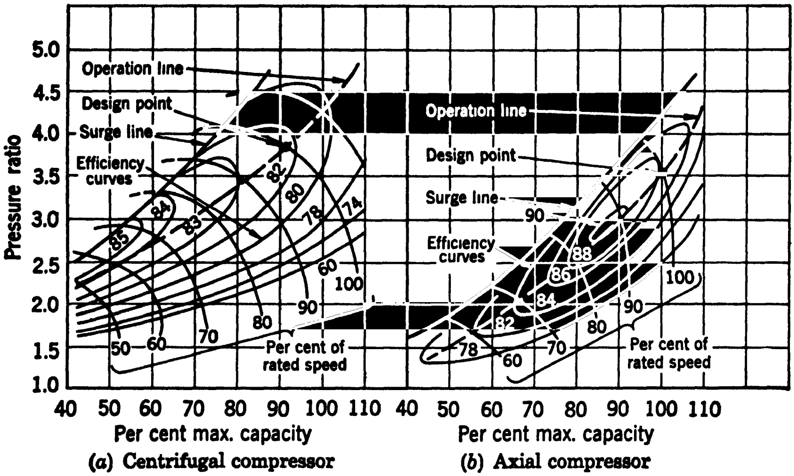


FIG. 231. Typical performance curves.

resistance in the system, it surges back in reverse flow until the pressure is relieved and it again moves forward. This reversal may occur

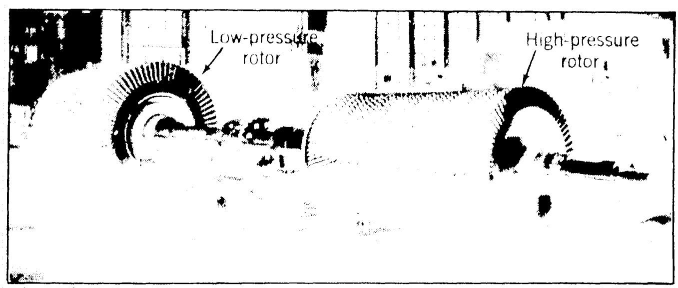


FIG. 232. Brown Boveri axial compressor rotors for a 10,000-kw gas-turbine installation.

at high frequency with consequent high stresses on the blower and must therefore be avoided.

260. Axial-Flow Compressors. Knowledge of air foils gained from wind tunnel studies has brought the axial compressor, Fig. 232, to the fore as the leading type for use with gas turbines. Isentropic efficiencies

of 85 to 90 per cent are attainable, the highest efficiencies being obtained with compressors of the larger sizes.

A full treatment of the theory of axial compressors is beyond the scope of this book, and some general statements must suffice. In the axial-flow compressor the blading consists of accurately designed air

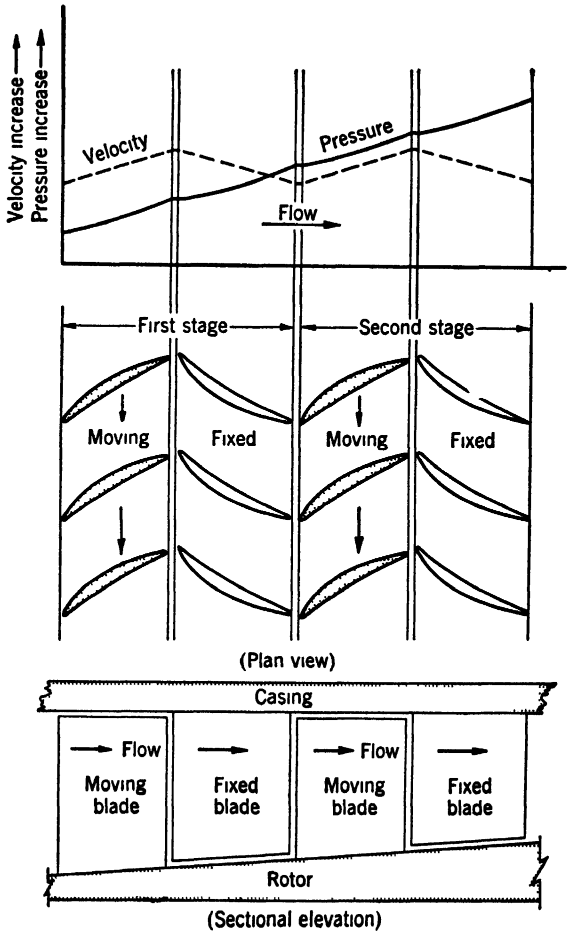


FIG. 233. Pressure-velocity relations in axial-flow compressor.

foils so mounted that on rotation the air is moved toward the trailing edge of the blades. The spaces between the blades or air foils are such that a diffusing and decelerating effect is accomplished as the air moves to the edge of the next set of moving blades. Properly shaped air foils are essential for high efficiency, for, as previously stated, it is much

easier to accelerate air efficiently than it is to decelerate it. A gradual change in size of the diffuser cross section tends to avoid turbulence. The blades must have the proper angle for air to enter with minimum shock and turbulence but the angle must be such that the blade is not easily stalled; i.e., the air stream may fail to follow the blade contour. The axial-flow compressor is more susceptible to stalling than the centrifugal compressor.

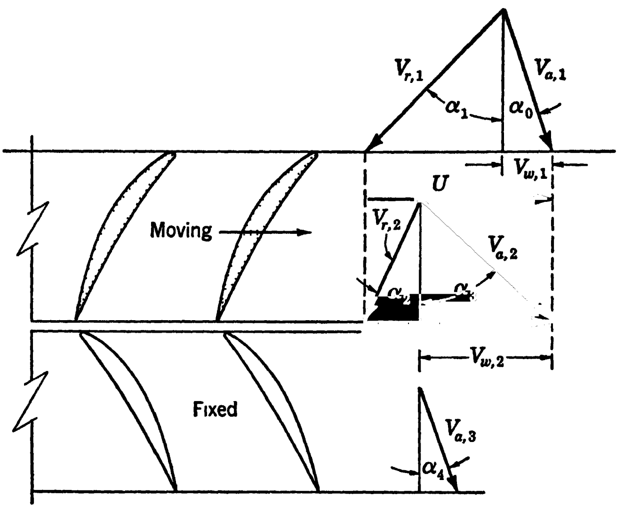


FIG. 234. Velocity diagram for axial-flow compressor.

In the moving blades the air acquires a rotational component which is redirected by the stationary elements. These stator elements can be as effective in obtaining pressure rise by diffusor action as are the moving blades. The stationary blades likewise redirect the air flow for proper entrance into the next set of moving blades. Figure 233 illustrates the pressure-velocity relations in an axial-flow compressor having symmetrical blading, in which equal work and enthalpy increases are obtained in the moving and fixed blades of a given stage. Figure 234 illustrates the velocity relations in which the air approaches the moving blade with absolute velocity $V_{a,1}$ and at an angle α_0 to the axis of rotation. The relative velocity $V_{r,1}$ is the resultant of $V_{a,1}$ and U (the blade velocity), and $V_{r,1}$ forms the angle α_1 with the compressor axis. After passing through the moving blades, the air emerges with a relative velocity, $V_{r,2}$, which is less than $V_{r,1}$ because of the compression accomplished by the moving blades. Angle α_2 is less than angle α_1 because of the blade shape. The resultant of $V_{r,2}$ and U is the ab-

solute velocity $V_{a,2}$ leaving the moving blade at angle α_3 with the axis. On entering the stationary blades the air is further diffused, decelerated, and on leaving is redirected to α_4, which is usually made equal to α_0. The air is now properly directed for entering the second stage.

The work done per pound of air per second is

$$W = \frac{U}{g}\,(V_{w,2} - V_{w,1}) \quad \text{ft-lb} \tag{265}$$

where U = blade velocity, fps.
$V_{w,1}$ = inlet whirl or tangential velocity, fps.
$V_{w,2}$ = outlet whirl or tangential velocity, fps.
$(V_{w,2} - V_{w,1})$ = the change in whirl produced by the rotor, fps.

All values for the above equation are taken at the mean radius of the rotor blades.

If the air should enter the rotor without prewhirl (obtained by inlet guide vanes), $V_{w,1}$ would be zero. Likewise for an ideal consideration the relative discharge velocity $V_{r,2}$ can be axial, in which case $V_{w,2} = U$ and equation 265 reduces to

$$W = \frac{U^2}{g} \quad \text{ft-lb for 1 lb of flow per sec} \tag{266}$$

Equation 266, it may be seen, is identical with equation 257 for the centrifugal compressor if no slip is assumed.

261. Efficiencies. *Compression efficiency* (e_c) is also called *isentropic* or *adiabatic efficiency* and is the ratio of the isentropic work to the actual indicated work.

$$e_c = \frac{\text{Isentropic work}}{\text{Actual indicated work}} \tag{267}$$

Thus the work of isentropic compression serves as a standard for compression work or power because a perfect machine with no heat transfer, no friction, and a perfect gas has been assumed.

Volumetric efficiency (e_v) of a reciprocating compressor may be obtained by actual measurement of air flow and displacement or by measurements on the indicator diagram. The result obtained by the method employing flow measurement is called *actual volumetric efficiency.* It is the ratio of the free air delivered by the compressor to the compressor displacement in cubic feet per minute; thus

$$e_v\ (\text{actual}) = \frac{\text{Free air delivered, cfm}}{\text{Displacement, cfm}} \tag{268}$$

$$e_v\ (\text{actual}) = \frac{\text{Mass of air delivered, lb}}{\text{Equivalent mass displaced, lb}} \qquad (269)$$

Using the indicator diagram, Fig. 222, the *apparent volumetric efficiency* is found from the ratio of the length 1 to 4 (scaled in inches) to the horizontal length 1 to 3 (scaled in inches). From Fig. 222

$$e_v\ (\text{apparent}) = \frac{V_1 - V_4}{V_1 - V_3} \qquad (270)$$

To calculate the volumetric efficiency in terms of clearance (c), Fig. 222, where

$$c = \frac{\text{Clearance volume}}{\text{Displacement}} = \frac{V_3}{V_1 - V_3}$$

for process 3 to 4

$$V_4 = V_3\left(\frac{P_3}{P_4}\right)^{1/k} = V_3\left(\frac{P_2}{P_1}\right)^{1/k} \quad \text{and} \quad V_3 = c(V_1 - V_3)$$

Therefore

$$V_4 = c(V_1 - V_3)\left(\frac{P_2}{P_1}\right)^{1/k}$$

Also

$$V_1 = (V_1 - V_3) + c(V_1 - V_3)$$

Therefore

$$e_v = \frac{V_1 - V_4}{V_1 - V_3} = \frac{(V_1 - V_3) + c(V_1 - V_3) - c(V_1 - V_3)\left(\frac{P_2}{P_1}\right)^{1/k}}{V_1 - V_3}$$

$$e_v = 1 + c - c\left(\frac{P_2}{P_1}\right)^{1/k} \qquad (271)$$

From equation 271 the volumetric efficiency can be calculated for isentropic compression when the clearance (c) is known. When the compression is polytropic, known or assumed values of the compression exponent (n) can be substituted for k. Clearance varies from 3 to 6 per cent, depending on the compressor size with large compressors having the smaller percentage.

If volumetric efficiency (e_v) is known, the free air delivered may be determined as the product of e_v and the displacement ($V_1 - V_3$). It follows that equation 244 may be written as

$$W = \frac{k}{1-k} P_1 e_v (V_1 - V_3)\left[\left(\frac{P_2}{P_1}\right)^{(k-1)/k} - 1\right] \qquad (272)$$

Volumetric efficiencies for compressors vary widely (60 to 90 per cent) and depend on intake resistance to the air flow, the clearance volume, and the discharge pressure. With a large clearance and high discharge pressure, re-expansion of the clearance air decreases the volumetric efficiency.

The *mechanical efficiency* of a compressor is the ratio of the total indicated horsepower of the air cylinders to the steam indicated horsepower, for a steam-driven unit. For a power-driven machine, the mechanical efficiency is the ratio of the air indicated horsepower to the brake horsepower of the power drive (turbine, motor, or belt). The *over-all* (sometimes called *compressor*) *efficiency* is the product of the compression efficiency and the mechanical efficiency and represents the ratio of the isentropic horsepower of the compressor to the horsepower supplied by the power drive.

262. Measurement of Air Flow. The quantity of air delivered by a blower or air compressor may be determined by one of the following: pitot tube, Venturi tube, thin-plate orifice, rounded orifice, short tube or nozzle, measuring tanks of known capacity, and recording meters. The results obtained may be expressed as free air, i.e., the volume at the temperature and pressure existing at the compressor or blower intake. The method employing the pitot tube is described in Art. 168. The Venturi tube is explained in Art. 249 and illustrated by Fig. 209.

The equation for the theoretical rate of flow of a perfect gas through a Venturi tube can be obtained by applying the steady-flow equation,

$$\frac{Z_1}{J} + \frac{P_1v_1}{J} + \frac{V_1^2}{2gJ} + u_1 + Q_1 = \frac{Z_2}{J} + \frac{P_2v_2}{J} + \frac{V_2^2}{2gJ} + u_2 + \frac{W}{J}$$

with the inlet as state point 1 and the throat as state point 2. When insignificant terms are eliminated

$$V_2 = \sqrt{2gJ(h_1 - h_2)} \tag{273}$$

Since

$$h_1 - h_2 = c_p(T_1 - T_2) \quad \text{or} \quad c_pT_1\left(1 - \frac{T_2}{T_1}\right)$$

and

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

it follows that

$$V_2 = \sqrt{2gJc_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{(k-1)/k}\right]} \tag{274}$$

where V_2 = theoretical velocity at throat, fps.
c_p = specific heat of constant pressure, Btu per (lb) (deg F abs).
P_1 = inlet pressure, psfa.
P_2 = throat pressure, psfa.
T_1 = temperature at inlet, deg F abs.
$k = c_p/c_v$ = isentropic coefficient = 1.4 for diatomic gases.

Assuming isentropic flow between inlet and throat,

$$P_2 v_2^k = P_1 v_1^k \quad \text{or} \quad v_2 = v_1 (P_1/P_2)^{1/k}$$

From $Pv = RT$, $v_1 = RT_1/P_1$ or

$$v_2 = \frac{RT_1}{P_1}\left(\frac{P_1}{P_2}\right)^{1/k} \quad \text{specific volume} \quad \text{cu ft per lb}$$

Substituting these values in the continuity equation, $m = A_1 V_1/v_1 = A_2 V_2/v_2$, Art. 30, equation 24,

$$m = \frac{223.8 A_2 \sqrt{c_p} P_1 C}{R\sqrt{T_1}} \sqrt{\left(\frac{P_2}{P_1}\right)^{2/k} - \left(\frac{P_2}{P_1}\right)^{(k+1)/k}} \quad \text{lb per sec} \tag{275}$$

Equation 275 is called the *Venturi* or *isentropic flow* equation for perfect gases and is applicable when

$P_2 > P_{critical}$, Art. 208.
R = the individual gas constant, ft-lb per (lb) (deg F).
C = coefficient of discharge for real nozzle.
A_2 = throat area, sq ft.

For a well-designed Venturi tube C has a value ranging between 0.98 and 0.94.

The *thin-plate* orifice in the simplest and most familiar form is a round hole in a thin, flat diaphragm (about 1/16 in. thick) clamped between the flanges at a point in a pipeline, the hole being concentric with the pipe; see Fig. 235. A thicker plate is sometimes used, and the plate is then chamfered around the hole on the downstream side, while the entrance side of the plate remains flat, with a sharp 90-deg corner at the edge of the opening. Pressure connections are provided in the pipe walls on both sides of the diaphragm so that pressure gages or manometers may be attached and the decrease of pressure through the orifice may be determined. This difference in pressure is utilized to find the rate of flow.

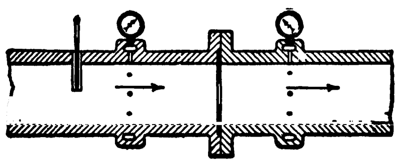


FIG. 235. Thin-plate-orifice installation.

The thin-plate orifice employs the Venturi principle, the orifice merely acting as a means whereby the fluid forms its own Venturi shape beyond the orifice plate. When the pressure connections are properly located the Venturi equation would apply, if the diameter of the minimum section (or vena contracta) of the air jet could be measured directly. Since

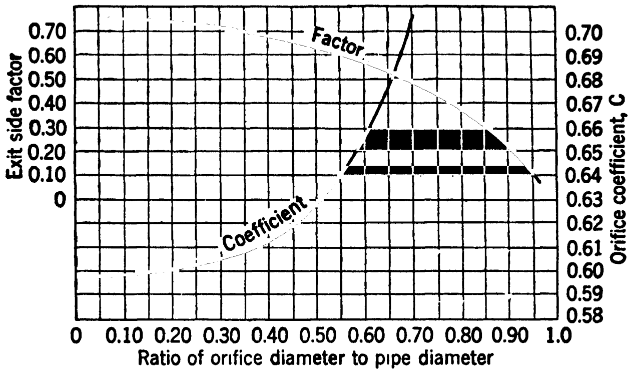


FIG. 236. Constants for thin-plate orifices.

the diameter of the vena contracta is not directly measurable it must, in practice, be replaced by the diameter of the orifice which can be measured. Experiments have shown that the position of minimum pressure, at the vena contracta, is geometrically similar in 3-in. to 12-in. pipes for the same orifice ratios. The pressure connection on the inlet side of the orifice is usually located one pipe diameter from the orifice. The distance from the orifice to the pressure connection, at the middle of the vena contracta, on the exit side may be determined by multiplying the pipe diameter by factors shown in Fig. 236.

The velocity of flow of air through a thin-plate orifice is

$$V = C\sqrt{2gH_a} \tag{276}$$

where V = velocity, fps.
C = orifice coefficient (see Fig. 236).
g = acceleration due to gravity, 32.17 ft per sec^2.
H_a = differential pressure, ft of air.

But

$$Q = \frac{AV}{144} \text{ cfs} \tag{277}$$

where

A = area of orifice sq in.

Hence

$$Q = \frac{\pi D^2}{144 \times 4} C\sqrt{64.34H_a} = 0.0438CD^2\sqrt{H_a} \tag{278}$$

However,

$$H_a = \frac{h_w \times d_w}{12 \times d_a} \tag{279}$$

Here h_w is the differential pressure across the orifice in inches of water; d_w and d_a are the densities in pounds per cubic foot of water in the manometer and of the air flowing, respectively. Therefore

$$Q = 0.0438CD^2\sqrt{\frac{d_w \times h_w}{d_a \times 12}}$$

and

$$Q = 0.0126CD^2\sqrt{h_w \frac{d_w}{d_a}}$$

but

$$h_w = \frac{1728p}{d_w}$$

thus

$$Q = 0.0126CD^2\sqrt{\frac{1728pd_w}{d_a d_w}}$$

$$Q = 0.526CD^2\sqrt{\frac{p}{d_a}} \tag{280}$$

where Q = air flowing through orifice, cu ft per sec at exit pressure and entrance temperature.
C = orifice coefficient from Fig. 236.
D = diameter of orifice, in.
p = observed differential pressure across orifice, psi.

The orifice coefficient varies according to the ratio of the orifice diameter as shown in Fig. 236. This coefficient represents the product of two factors, a frictional coefficient which allows for the frictional resistance at the entrance to the orifice and the contraction of the stream flow which is the ratio of the area of the vena contracta to the area of the orifice.

Example. Air is flowing through a ¾-in. thin-plate orifice, the latter being inserted between the flanges of a 3-in. pipe. The pressure on the upstream side (3 in. from orifice) is 20 in. of mercury above atmosphere, and the pressure on the

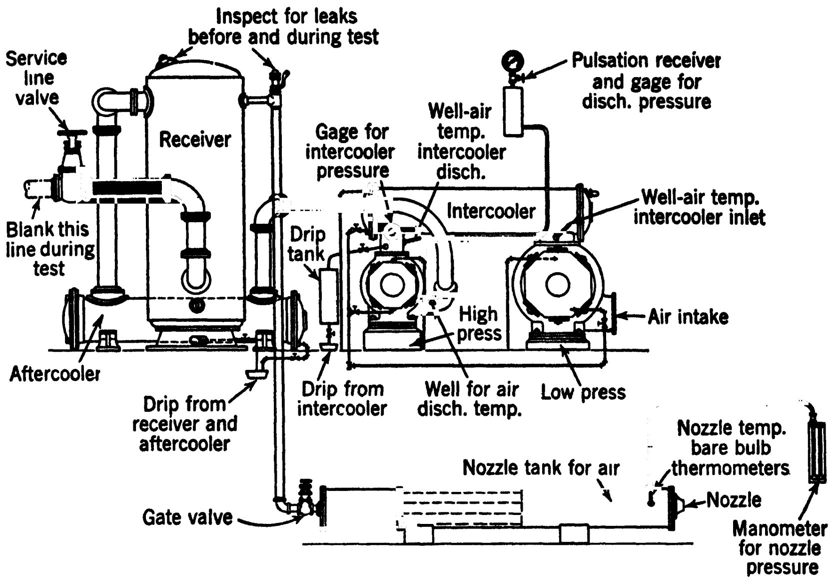


FIG. 237. Test arrangement for air measurement by low-pressure nozzle.

downstream side (at the vena contracta, $2\frac{3}{16}$ in. from orifice) is 4 in. of mercury. Find the cubic feet per minute of air, if the temperature of the air on the upstream side is 140 F and the barometer is 29.32 in. mercury.

Solution.

$$p = (20 - 4) \times 0.49 = 7.84 \text{ psi}$$

$$C = 0.603, \text{ from curve (Fig. 236)}$$

$$d_a = \frac{m}{V} = \frac{P}{RT} = \frac{(4 + 29.32) \times 0.49 \times 144}{53.35 \times 600}$$

$$d_a = 0.0735 \text{ lb per cu ft}$$

Substituting in equation 280 gives

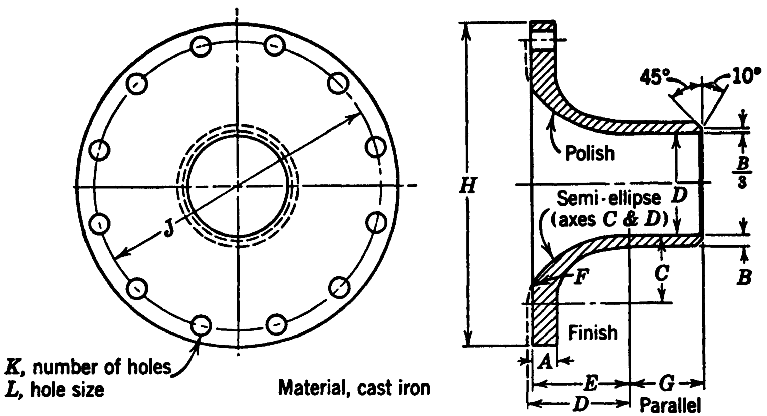
$$Q = 0.526 \times 0.603 \times \overline{0.75}^2 \sqrt{\frac{7.84}{0.0735}}$$

$$Q = 0.18 \times 10.34 = 1.845 \text{ cfs}$$

$$= 110.7 \text{ cfm at 4 in. of mercury and 140 F}$$

The *low-pressure nozzle* test is recommended by the ASME and the Compressed Air and Gas Institute as an accurate method for measuring the air output of a compressor. The air delivered by the compressor is discharged into the receiver, Fig. 237, where the desired pressure is maintained constant by bleeding air through a gate valve to a tank provided with a nozzle. The diameter of the nozzle tank should be at

least four times the nozzle diameter, preferably larger; the minimum length of this tank should be forty times the nozzle diameter. The exit nozzle is merely an orifice open to the atmosphere with curved inner edges and is of such size that the pressures in the nozzle tank will



D in.	A in.	B in.	C in.	E in.	F in.	G in.	H in.	J in.	K	L in.	Flow Rates cfm	
											Min.	Max.
1.000	0.9375	.0250	0.67	0.969	0.05	0.875	9.00	7.50	8	0.750	62	127
1.375	1.000	0.250	0.92	1.332	0.07	1.063	11.00	9.50	8	0.875	119	239
2.000	1.000	0.313	1.33	1.938	0.10	1.500	11.00	9.50	8	0.875	253	506
2.500	1.000	0.375	1.67	2.422	0.13	1.875	11.00	9.50	8	0.875	397	790
3.000	1.000	0.375	2.00	2.906	0.15	2.250	11.00	9.50	8	0.875	565	1,127
4.000	1.125	0.438	2.67	3.875	0.20	3.000	13.50	11.75	8	0.875	1,010	2,020
5.000	1.188	0.500	3.33	4.844	0.25	3.750	16.00	14.25	12	1.000	1,590	3,160
6.000	1.250	0.500	4.00	5.812	0.30	4.500	19.00	17.00	12	1.000	2,260	4,510
8.000	1.438	0.625	5.33	7.750	0.40	6.000	23.50	21.25	16	1.125	4,050	8,100
10.000	1.688	0.625	6.67	9.688	0.50	7.500	27.50	25.00	20	1.250	6,350	12,600
12.000	1.875	0.750	8.00	11.625	0.60	9.000	32.00	29.50	20	1.375	9,100	18,200

FIG. 238. ASME long-radius, low-pressure, air-nozzle details.

not be more than $1\frac{1}{2}$ psig above atmosphere, when passing all the air through its opening. The exact pressure in the nozzle (discharge) tank is determined by a water manometer, and the temperature of the air flowing is measured by a thermometer passing through the tank wall, about four times the nozzle diameter from the nozzle. From these observations and the barometric reading, the cubic feet of air flowing through the nozzle per second may be determined. Figure 238 shows a drawing of the low-pressure nozzle with an accompanying table of dimensions and capacities as recommended by the ASME. The inner surface of the nozzle must be highly polished, giving a nozzle coefficient of 0.98 to 0.99, depending upon the shape and finish.

When the nozzle, as selected by the table given in Fig. 238, is of such size that the manometer reading is from 10 to 40 in. of water, the quantity of air flowing through the nozzle may be calculated by the equation

$$Q = 0.32KD^2T_2\sqrt{\frac{(p_1 - p_2)}{p_2T_1}} \tag{281}$$

where Q = cubic feet of air flowing per second at observed pressure and temperature of downstream side of nozzle.
K = nozzle coefficient, 0.98 to 0.99.
D = diameter of smallest part of nozzle throat, in.
T_1 = absolute F temperature, upstream side of nozzle.
T_2 = absolute F temperature, downstream side of nozzle.
p_1 = pressure, upstream side of nozzle, psia.
p_2 = pressure, downstream side of nozzle, psia.

Example. The air discharged from an air compressor flows through a 4-in. ASME low-pressure nozzle into the atmosphere. The observed manometer reading is 12 in., the temperature of the air entering the nozzle is 206 F, and the barometer is 29.40 in. of mercury. How many cubic feet of air are being compressed per minute if the nozzle is of such shape and finish that its coefficient is 0.99? Temperature of the atmosphere, 80 F.

Solution. $p_1 - p_2 = \frac{12}{13.6} \times 0.491 = 0.433$ psi

$$p_2 = 29.40 \times 0.491 = 14.44 \text{ psia}$$

Substituting in equation 281 gives

$$Q = 0.32 \times 0.99 \times 16 \times 540\sqrt{\frac{0.433}{14.44 \times 666}}$$

$$Q = 18.4 \text{ cfs}$$

$$= 1104 \text{ cfm at 80 F and 29.4 in. of mercury}$$

263. Testing Air Compressors. Most air compressors are driven at constant speed by the driving engine, and in testing such compressors the primary variable is the receiver pressure; see Fig. 237. This pressure may be controlled by the rate at which the air is discharged from the receiver. Observations to be made are (1) receiver pressure, (2) speed of compressor, (3) inlet conditions (temperature and barometric pressure) of the free air, (4) temperature and pressure at the inlet to the measuring orifice, (5) input power, and (6) indicated power if an indicator can be used. If the unit is a two-stage machine, the temperatures to and from the intercooler and the pressure within the intercooler should also be recorded.

Both the indicated and the shaft horsepower will be found to increase

directly as the discharge pressure is increased. The volumetric efficiency will decrease with increased discharge pressure, owing to the leakage and the effect of the clearance volume.

PROBLEMS

1. A horizontal double-acting piston-type compressor has a 24-in. bore and a 24-in. stroke. A 3-in.-diameter piston rod and tail rod extend through the cylinder heads. Determine the displacement in cubic feet per minute if the compressor operates at 200 rpm.

2. Find the isentropic air horsepower required to compress and deliver 500 cfm of free air from 14.696 psia and 60 F to a receiver at 100 psia. 81.8 hp, *Ans.*

3. Ten pounds of air per minute are to be delivered at 80 psia. Inlet conditions are 14.696 psia and 60 F. Assuming 100 per cent volumetric efficiency and neglecting clearance, determine the size of a single-acting compressor operating at 160 rpm, if the bore and stroke are equal.

4. Using the conditions given in problem 3, find (*a*) the isentropic air horsepower, (*b*) the horsepower with isothermal compression, and (*c*) the heat that must be rejected in (*a*) and (*b*).

5. Find the shaft horsepower necessary to compress 500 cfm of air from 14.7 psia and 60 F to 100 psia in a single-stage reciprocating compressor if the compression efficiency (isentropic) is 90 per cent and the mechanical efficiency is 80 per cent. 113.6 shp, *Ans.*

6. Six pounds of air per minute are to be delivered at 300 psia from initial conditions of 14.696 psia and 80 F. If a two-stage compressor is used, find (*a*) the best intercooler pressure, assuming isentropic compression and intercooling to the original temperature; and (*b*) the Btu per hour to be removed in the intercooler.

7. Using the conditions in problem 6 find (*a*) the horsepower required and (*b*) the shaft horsepower if the isentropic compression efficiency is 90 per cent and the mechanical efficiency is 80 per cent.

8. (*a*) Find the horsepower required to compress 6 lb of air per min from 14.696 psia to 300 psia in a two-stage compressor with perfect intercooling. Clearance, 5 per cent; $n = 1.20$ for compression and re-expansion; inlet air temperature, 80 F. (*b*) Calculate the heat removed by the intercooler, in Btu per hour.

9. A centrifugal compressor is to deliver 1 lb of air per sec. The tip speed of the impeller is 1200 fps; the slip factor is 0.9; the power input factor is 1.04. Find the shaft horsepower developed. 76.0 shp, *Ans.*

10. Find the discharge pressure that may be expected from a single-stage centrifugal compressor operating at a blade-tip speed of 1200 fps if the isentropic efficiency is 85 per cent, the slip factor is 0.9, the power input factor 1 04, the inlet stagnation temperature 520 F, the inlet stagnation pressure 14.696 psia, and the mean specific heat 0.25. Use air as medium, $k = 1.40$.

11. In a centrifugal compressor the inlet stagnation conditions are 510 F abs and 14.2 psia. The outlet stagnation pressure is 142 psia and the isentropic efficiency is 82 per cent. Find the outlet stagnation temperature. Illustrate the compression path on a *Ts* diagram. Use air as medium, $k = 1.40$.

12. A rotary compressor receives superheated ammonia at 20 F and discharges it in steady flow to a condenser at 230 F. Heat transfer and velocity effects are negligible. If the mechanical efficiency is 90 per cent and mean specific heat is 0.45 Btu per lb, deg F, find the shaft horsepower required for a flow of 10 lb per min. 24.78 shp, *Ans.*

13. A single-stage single-acting 12-in. bore by 18-in. stroke piston-type compressor has 6 per cent clearance volume and discharges at 84 psia with intake at 80 F and 14.0 psia. Assuming isentropic conditions on compression and re-expansion of the clearance air, find (*a*) the volumetric efficiency, per cent; (*b*) the free air delivered per minute when operating at 200 rpm; and (*c*) the shaft horsepower if the over-all efficiency is 85 per cent.

14. A single-stage double-acting air compressor has a mechanical efficiency of 80 per cent and a clearance volume of 10 per cent. If 100 lb per hr of free air are compressed isentropically from 14.7 psia and 60 F to 102.9 psia, what shaft horsepower is required?

15. Air flowing in a 6-in. I.D. pipe is measured by a thin-plate orifice of 3.0-in. diameter. The differential pressure across the orifice is 1.5 psi. Find the weight of the air flow in pounds per second if the downstream pressure is 30 psia and the entrance temperature is 60 F. 9.21 cfs, *Ans.*

16. A Venturi has a 3-in. diameter inlet and receives air at 100 psia and 120 F. The exit pressure is 70 psia. If the throat diameter is 1.2 in. and the Venturi coefficient $C = 0.97$, find the air flow in pounds per minute.

CHAPTER 15

Gas Turbines

264. Introduction. The simple gas turbine, Fig. 239, consists of an *air compressor*, a *combustor*, a *turbine*, and several auxiliaries as influenced by weight-power ratio and speed characteristics. The auxiliaries must provide for lubrication, speed control, fuel supply, and starting.

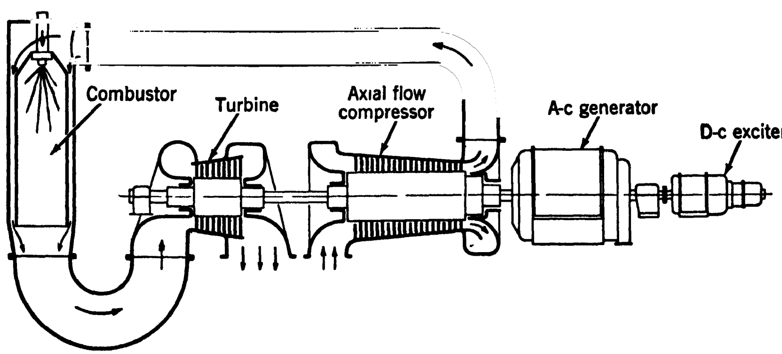


FIG. 239. Simple-cycle gas turbine with necessary auxiliaries.

Several practical limitations are encountered in gas-turbine operation; such limitations largely determine the maximum performance that can be obtained. Among these are blade temperature, blade speed, compressor efficiency, turbine efficiency, and heat transfer (in regenerative cycles).

In operation of the simple gas turbine, compressed air is delivered to the combustor where fuel is supplied in steady flow and a continuous flame is maintained, the original ignition usually being made by means of a spark. The air, heated in the combustor, expands through nozzles and develops a high velocity. Some of the kinetic energy of the air stream is delivered to the blades of the turbine. Part of this energy is used to drive the compressor, and the remainder is available for shaft work. Like steam-turbine operation, the whole process is a typical example of steady flow—the primary difference being that air, which

is a relatively perfect gas, is used in place of steam or other condensable media. Close similarities are found between the gas turbine and the steam turbine since the underlying thermodynamic principles are much the same.

In the open system, Fig. 239, the products of combustion flow with the air stream through the turbine. A high percentage of excess air (300 to 600 per cent) is necessary in order to dilute the combustion products to a temperature that the turbine wheel can withstand (1200 to 1800 F). In the closed system the products of combustion do not pass through the turbine but are directed through a heat exchanger. The gases passing through the turbine are made to operate in a closed circuit, being consecutively compressed, heated, expanded, and cooled. The closed system permits the use of fuel of any type in the combustor but a heat exchanger is required. Therefore the closed system is limited to stationary plants.

Of the many advantages in gas turbines over vapor systems, a few are (1) a more compact system, (2) fewer auxiliaries, (3) no condenser maintenance, (4) no water requirements, (5) simple lubrication system, (6) ease of control, (7) light foundation, (8) clean exhaust (no stack required), and (9) low weight-power ratio. Some of the advantages of gas turbines over reciprocating engines for aircraft are (1) use of kerosene or distillate, (2) less fire hazard, (3) no unbalanced forces, (4) simpler cooling problems, (5) fewer moving parts, (6) easy installation, (7) not being limited by propeller characteristics, (8) smaller frontal area, and (9) less weight per horsepower.

265. Applications. Aside from its use in aircraft, the gas turbine has its widest application as a pumping engine for pipelines carrying petroleum products since its continuous combustion process permits the use of all grades of oil, gas, and gasoline as fuel, the only limitation on a fuel being that the products of combustion must not cause either abrasion or deposits. The successful use of pulverized coal as a fuel would greatly accelerate the use of the gas turbine for power generation. At the present time research is directed toward the problems of blade erosion and deposits caused by the presence of ash in coal.[1] The cycle efficiency will improve as blade materials which will withstand higher temperatures are discovered and methods for cooling the blades are developed. It is important that the highest possible compressor efficiency be obtained since large quantities of air must be handled and since the compressor absorbs approximately three fourths of the turbine output. A 1000-hp unit may actually generate 4000 hp, three

[1] Test results for a 4250-hp locomotive-type coal-burning gas turbine are given in *Power Engineering*, May, 1952, p. 56.

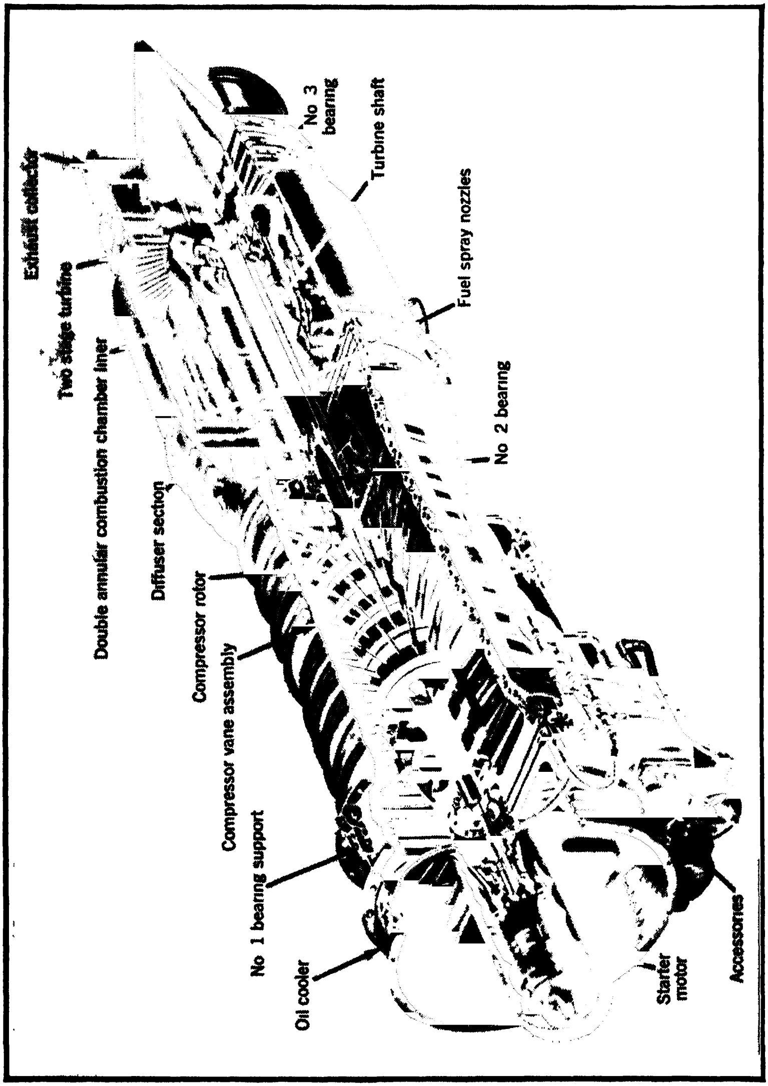


FIG. 240 Westinghouse J34 turbojet engine.

thousand of which would be used by the compressor. Power outputs as high as 27,000 kw and over-all efficiencies of 35 per cent have been obtained.

For jet engines the useful output results from the acceleration given to the mass flow of air. The product of this mass and the acceleration

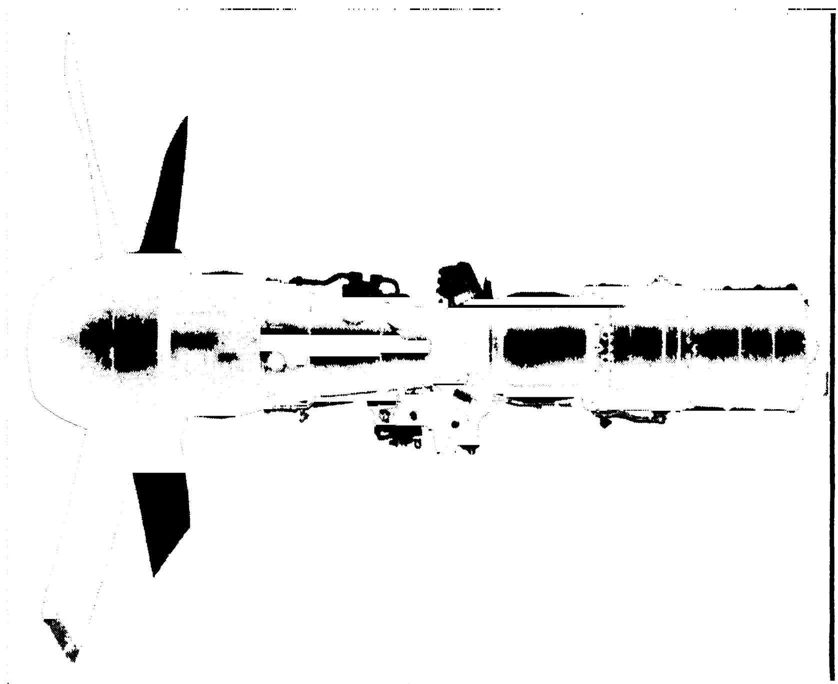


FIG 241 Exterior view of Pratt and Whitney Aircraft T-34 Turbo-Wasp engine.

is equal to the total force or thrust developed Figure 240 shows a cutaway view of a J34 engine. At 12,500 rpm this engine develops a stable thrust of 3000 lb It is essentially 2 ft in diameter and 10 ft long, having two turbine stages and eleven compressor stages. The compression ratio is approximately 4.1. The gross weight including accessories is 1200 lb. The fuel is light oil, kerosene, or gasoline, and the specific fuel consumption is 1 08 lb per bhp-hr.

The high specific fuel consumption of the gas turbine is the primary objection to its wider use This problem is being met, however, by use of multistage compression, multistage expansion, and regeneration.

Figure 241 shows a turboprop engine of axial-flow type. This engine develops a total of 5700 hp. Approximately 90 per cent of the

power is delivered to the propeller, and 10 per cent is developed by the jet. The basic engine weight is 2550 lb, giving a ratio of 1 hp for each 0.448 lb of engine. The specific fuel consumption of 0.62 lb per bhp-hr compares favorably with that of large piston-type spark-ignited engines.

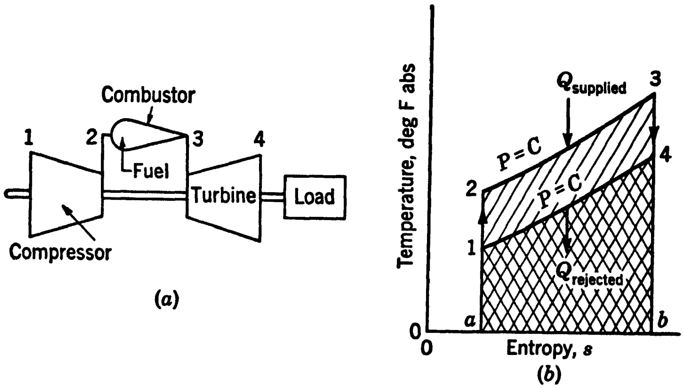


FIG. 242. Simple gas-turbine cycle.

266. Simple Cycle. In 1873, Bayton conceived the idea of the present simple gas-turbine cycle which is the basis for gas-turbine analysis. Inspection of Fig. 242 shows the cycle consisting of isentropic compression (1 to 2), constant-pressure energy addition (2 to 3), isentropic expansion (3 to 4), and constant-pressure energy release (4 to 1). As thus defined, it is a reversible cycle for which the efficiency would be

$$e = \frac{Q_{supplied} - Q_{rejected}}{Q_{supplied}} = \frac{mc_p(T_3 - T_2) - mc_p(T_4 - T_1)}{mc_p(T_3 - T_2)} \qquad (282)$$

$$e = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1}{T_2}\left(\frac{\frac{T_4}{T_1} - 1}{\frac{T_3}{T_2} - 1}\right)$$

but

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = \left(\frac{V_4}{V_3}\right)^{k-1} = \frac{T_3}{T_4}$$

Therefore

$$\frac{T_4}{T_1} = \frac{T_3}{T_2}$$

or

$$e = 1 - \frac{T_1}{T_2} \quad \text{(same as for Otto cycle)} \tag{283}$$

$$e = 1 - \left(\frac{V_2}{V_1}\right)^{k-1} \tag{284}$$

$$e = 1 - \frac{1}{\left(\frac{P_2}{P_1}\right)^{(k-1)/k}} \tag{285}$$

where T = temperature, deg F abs.
P = pressure, psfa.
V = volume, cu ft.
k = 1.4.
c_p = specific heat of constant pressure, Btu per lb.
m = mass flow of air, lb.
Subscripts refer to state points in Fig. 242.

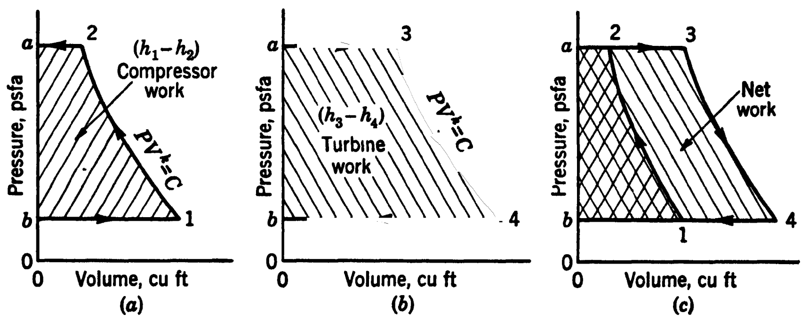


FIG. 243. Ideal diagram for simple gas turbine.

The expression P_2/P_1 is called *pressure ratio*. Equation 285 is the most useful form.

Figure 243(a) shows the ideal compressor work, (b) the ideal turbine work, and (c) the net difference. Thus, *for the compressor*

$$W = {}_bW_1 + {}_1W_2 + {}_2W_a + {}_aW_b$$

or

$$W = P_1(V_1 - V_b) + \frac{P_2V_2 - P_1V_1}{1 - k} + P_2(V_a - V_2) + 0$$

Since $V_a = 0$ and $V_b = 0$, it follows that

$$W = \frac{k}{1-k}(P_2V_2 - P_1V_1) \quad \text{ft-lb} \tag{286}$$

$$\frac{W}{J} = c_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{(k-1)/k}\right] \quad \text{Btu per lb} \tag{287}$$

$$\frac{W}{J} = h_1 - h_2 = c_p(T_1 - T_2) \quad \text{Btu per lb} \tag{288}$$

where W = work, ft-lb per lb of air flow.
h = enthalpy, Btu per lb of air flow.
Other terms as defined previously.

For the turbine

$$W = {}_bW_a + {}_aW_3 + {}_3W_4 + {}_4W_b$$

$$W = 0 + P_3(V_3 - V_a) + \frac{P_4V_4 - P_3V_3}{1 - k} + P_4(V_b - V_4)$$

Since $V_a = 0$ and $V_b = 0$, it follows that

$$W = P_3V_3 + \frac{P_4V_4 - P_3V_3}{1 - k} - P_4V_4$$

$$W = \frac{k}{1 - k}(P_4V_4 - P_3V_3) \quad \text{ft-lb} \tag{289}$$

$$\frac{W}{J} = c_pT_3\left[1 - \left(\frac{P_1}{P_2}\right)^{(k-1)/k}\right] \quad \text{Btu per lb} \tag{290}$$

$$\frac{W}{J} = h_3 - h_4 = c_p(T_3 - T_4) \quad \text{Btu per lb} \tag{291}$$

where all terms are as defined previously.

The net work of the ideal cycle is the algebraic sum of the compressor and turbine work. Expressions for compressor work give negative values, and those for turbine work give positive values; thus the sum of the work around the cycle 1–2–3–4–1 can be obtained from several combinations.

Example. An ideal gas turbine has a pressure ratio of 4 : 1 and uses 50 lb of air per sec. Intake pressure and temperature are 14.7 psia and 70 F. Assume air as the only medium with constant $c_p = 0.24$ and $k = 1.4$. Maximum temperature is 1500 F. Find (*a*) the cycle efficiency, (*b*) the net work per second, and (*c*) the theoretical horsepower.

Solution.

(*a*)
$$e = 1 - \frac{1}{\left(\frac{P_2}{P_1}\right)^{(k-1)/k}} = 1 - \frac{1}{(4)^{0.4/1.4}}$$

$$= 0.3275, \quad \text{or} \quad 32.75 \text{ per cent}$$

(*b*)

Net work = algebraic sum of compressor and turbine work

$$\frac{W}{J} = c_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{(k-1)/k}\right] + c_pT_3\left[1 - \left(\frac{P_1}{P_2}\right)^{(k-1)/k}\right] \text{ Btu per lb}$$

$$= 0.24 \times 530(1 - 4^{0.286}) + 0.24 \times 1960(1 - 0.25^{0.286})$$

$$= 92.2 \text{ Btu per lb}$$

$$W = 92.2 \times 50 \times 778.26 = 3{,}590{,}000 \text{ ft-lb per sec}$$

(*c*)

$$\text{Power} = \frac{3{,}590{,}000}{550} = 6530 \text{ hp}$$

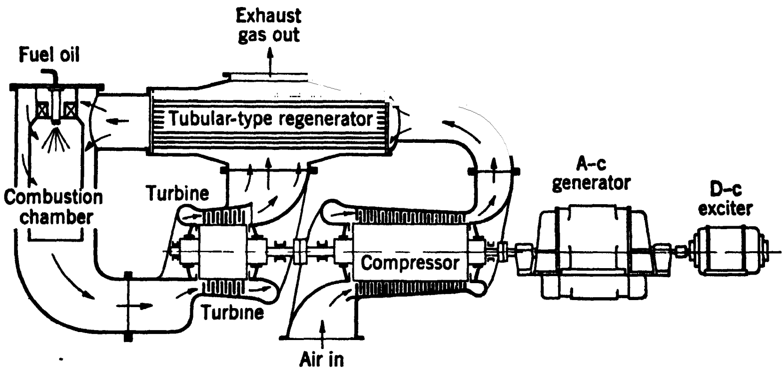


FIG. 244. Brown Boveri regenerative gas turbine.

267. Regenerators. Recovering waste heat from the high-temperature exhaust gases of a gas turbine is a means of improving the cycle efficiency. In steam power plants an air preheater, Art. 122, extracts energy from the stack gases to heat the combustion air. Heat exchangers or regenerators for gas turbines are of the same type, i.e., tubular or rotary-plate. In the tubular type, which is more widely used, the exhaust gases are made to flow through the tubes, and the air to be heated is forced across the outside of the tubes. The rotary type can be built in a more compact form, and true counterflow heat transfer is easily obtained; see Fig. 85. Although this type is more compact, its complicated construction limits its application.

The use of a regenerator in a gas turbine presents three problems: necessity of large area of tube surface, maintenance of clean surfaces, and increased resistance to flow. In power plants the large bulk of a heat exchanger can be accommodated, and mechanical soot blowers, Fig. 63, can be used to clean the tube surfaces. However, the bulk of a

heat exchanger makes it impractical for gas turbines as used in aircraft or locomotives.

Figure 244 shows a single-shaft unit equipped with a tubular-type heat exchanger. This type of unit is built in capacities as high as 8000 kw.

An energy balance for a regenerator may be obtained by equating all the energy received by the cold medium to that extracted from the hot medium. Assuming no heat losses and no leakage, from Fig. 245(*b*),

$$m_a c_{pa}(T_{a,2} - T_{a,e}) = m_g c_{pg}(T_{g,4} - T_{g,f})$$

or

$$m_a(h_{a,2} - h_{a,e}) = m_g(h_{g,4} - h_{g,f}) \tag{292}$$

where m = the weight flow, lb per sec.
a = subscript designating air.
g = subscript designating exhaust gas.
c_p = specific heat of constant pressure, Btu per lb.
T = temperature, deg F abs.
h = enthalpy, Btu per lb.
Other subscripts as in Fig. 245(*b*).

The specific heat of the gases will be slightly greater than that of the air, owing to the higher gas temperature and the effect of the products of combustion. Likewise the weight of gas flow will be greater than the weight of the air flow by the amount of fuel burned; therefore, the rise in air temperature will be greater than the drop in gas temperature, although the effect of stray heat losses tends to make them approximately equal.

The energy extracted from the exhaust gases and absorbed by the air can also be expressed by

$$Q = UA\theta_m \tag{293}$$

where Q = heat transferred, Btu per hr.
U = over-all coefficient of heat transfer, Btu per (hr) (sq ft) (deg F).
A = area of heat transfer surface, sq ft.
θ_m = mean temperature difference, deg F.

The logarithmic mean temperature difference, Art. 233, is usually used for θ_m, where

$$\theta_m = \frac{\theta_a - \theta_b}{\log_e \dfrac{\theta_a}{\theta_b}}$$

where θ_a = difference in temperature between incoming hot fluid and its surroundings, deg F.

θ_b = difference in temperature between outgoing hot fluid and its surroundings, deg F.

Values of the over-all coefficient U vary widely and are dependent on the following factors: (1) gas velocity, (2) tube diameter, (3) flow pattern, (4) conductivity and thickness of tube material, as well as certain properties of the gases between which the heat is being transferred. Calculation of U with reasonable accuracy can be made by heat transfer theory, but more often U is determined experimentally. Values of U for a tubular heat exchanger range from 2.0 to 6.0 Btu per (hr) (sq ft) (deg F).

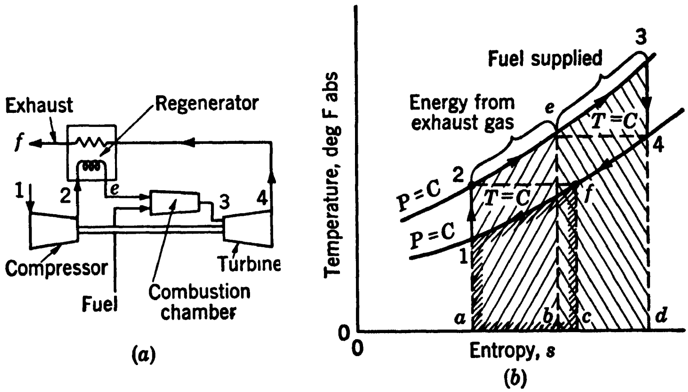


FIG. 245. Ideal regenerative Brayton cycle.

The *effectiveness of a regenerator* in transferring all the energy available from the exhaust gases to the air which is to be heated is expressed as

$$e_e = \frac{\text{Rise in air temperature}}{\text{Maximum possible rise}}$$

or

$$e_e = \frac{T_e - T_2}{T_4 - T_2} \tag{294}$$

where T = temperature, deg F abs. The subscripts refer to state points in the cycle diagram of Figs. 245(b) and 246.

268. Ideal Regenerative Cycle. The efficiency of the simple Brayton cycle can be greatly improved by the use of regeneration, i.e., the use of exhaust energy to heat the compressed air before it enters the combustion chamber. Such an arrangement is shown in Fig. 245.

The advantage of regeneration may be observed in Fig. 245(*b*), where it is apparent that the gases leaving the compressor at point 2 may be heated up to point *e* which is at the same temperature as the exhaust gases leaving the turbine at point 4. In Fig. 245(*b*) the only energy to be discarded is shown as area *a*–1–*f*–*c*–*a*. It is also evident that the heat absorbed by the air in the heat exchanger is equal to that taken

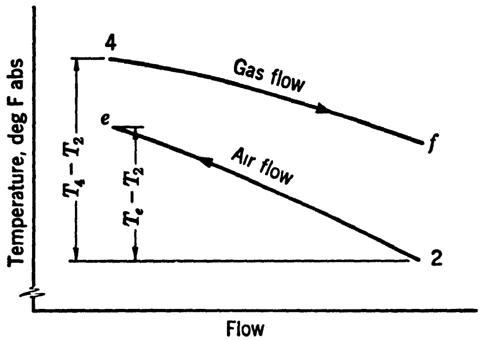


FIG. 246 Regenerator effectiveness

from the exhaust gases; therefore, area *a*–2–*e*–*b*–*a* equals area *c*–*f*–4–*d*–*c* and represents energy exchanged per pound of air flowing for the ideal case. The net work is area 1–2–3–4–1 as for the simple cycle, and the net energy supplied is *b*–*e*–3–*d*–*b* per pound of air.

The ideal regenerative cycle efficiency is

$$e_r = \frac{Q_{supplied} - Q_{rejected}}{Q_{supplied}}$$

$$e_r = \frac{mc_p(T_3 - T_e) - mc_p(T_f - T_1)}{mc_p(T_3 - T_e)}$$

$$e_r = 1 - \frac{T_f - T_1}{T_3 - T_e} \tag{295}$$

But $T_f = T_2$ and $T_e = T_4$; therefore

$$e_r = 1 - \frac{T_1}{T_3}\left(\frac{\dfrac{T_2}{T_1} - 1}{1 - \dfrac{T_4}{T_3}}\right)$$

Also $T_4/T_3 = T_1/T_2$; therefore

$$e_r = 1 - \frac{T_1}{T_3}\left(\frac{\dfrac{T_2 - T_1}{T_1}}{\dfrac{T_2 - T_1}{T_2}}\right) \quad \text{Fig. 245} \tag{296}$$

Since $T_2/T_1 = (P_2/P_1)^{(k-1)/k}$, it follows that

$$e_r = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{(k-1)/k} \tag{297}$$

where T = temperature, deg F abs.
P = pressure, psia.
k = 1.4

The subscripts refer to state points in Fig. 245.

From equation 297 it is apparent that the ideal regenerative efficiency is a function of the pressure ratio P_2/P_1 and the temperature ratio T_1/T_3. Since P_1, T_1, and T_3 are usually fixed by circumstance, it is interesting to note that e_r approaches the Carnot efficiency $(1 - T_1/T_3)$ as P_2 approaches P_1; also that e_r approaches zero when P_2 becomes infinitely large.

Example. An ideal regenerative gas turbine has a pressure ratio of 4 to 1 and uses 50 lb of air per sec. Intake pressure and temperature are 14.7 psia and 70 F. Assume air is the only medium with c_p constant at 0.24 and $k = 1.4$. Maximum temperature of the cycle is 1500 F. Find (*a*) the cycle efficiency, (*b*) the net work per second, and (*c*) the theoretical horsepower.

Solution.

(*a*)
$$e_r = 1 - \frac{T_1}{T_3}\left(\frac{P_2}{P_1}\right)^{(k-1)/k}$$

$$T_1 = 70 + 460 = 530 \text{ F abs}$$

$$T_3 = 1500 + 460 = 1960 \text{ F abs}$$

$$\frac{P_2}{P_1} = 4$$

$$e_r = 1 - \frac{530}{1960} \times 4^{0.286} = 0.598, \quad \text{or} \quad 59.8 \text{ per cent}$$

(*b*) Net work, as for the simple cycle,

$$\frac{W}{J} = c_pT_1\left[1 - \left(\frac{P_2}{P_1}\right)^{(k-1)/k}\right] + c_pT_3\left[1 - \left(\frac{P_1}{P_2}\right)^{(k-1)/k}\right]$$

$$\frac{W}{J} = 0.24 \times 530[1 - (4)^{0.286}] + 0.24 \times 1960[1 - (0.25)^{0.286}]$$

$$\frac{W}{J} = 92.2 \text{ Btu per lb}$$

$$W = 92.2 \times 50 \times 778 = 3{,}590{,}000 \text{ ft-lb per sec}$$

(*c*) $$\text{Power} = 3{,}590{,}000 \div 550 = 6530 \text{ hp}$$

From the above example it is apparent that ideal efficiency may be greatly improved by regeneration while the work per pound of air flowing remains constant. Also, observe from equation 297 that greatest efficiency with regeneration is obtained with the lowest compression ratio. Practical considerations determine the limit to which these trends may be followed.

269. Deviations from the Ideal Cycle. The actual cycle efficiency and power are less than the ideal for various reasons, some of which are:

1. Air is not a perfect gas.
2. The medium is not pure air since fuel has been added.
3. The quantity of the medium may change because of leakage, fuel injection, and bleed air used for cooling.
4. Compression is not isentropic.
5. Expansion is not isentropic.
6. Pressure losses occur throughout the system.
7. Not all energy supplied in fuel is released on combustion.
8. Energy escapes from the system (nonadiabatic).
9. Exhaust velocity is not used except in jet aircraft.

The increase in specific heat of air with an increase in temperature was discussed in Art. 37. In open cycles the fuel is added to the air stream, and, although this has the advantage of increasing the mass flow through the turbine, it has the disadvantage of increasing the mean specific heat of the medium, which in turn requires more energy to raise the medium to a high-energy level. Although some leakage occurs, such as end flow past the turbine and compressor blades, the effect is not serious. However, in certain systems employing a rotating regenerative-type heater in which a matrix or labyrinth is moved alternately from the gas stream to the air stream, the leakage and carry-over may be 2 or 3 per cent of the total air flow.

The theory of compressors has been discussed in Chapter 14, from which it will be recalled that all compressors fall far short of the isentropic ideal. Currently the best designs give isentropic efficiencies for axial-flow compressors at 87 to 90 per cent and for centrifugal compressors at 75 to 85 per cent.

Pressure losses in the duct system may seriously impair the efficiency and power unless every means is taken to minimize the flow resistance in the system. Incomplete combustion usually claims 1 or 2 per cent of the energy supplied because of the high rate at which the air must pass through the burner. Radiation losses are small. Certain parts such as turbine disks, blades, nozzles, and combustion chamber walls are usually cooled by bleeding air from the compressor. This action produces a loss in heat energy as well as in the work of compression. Although mechanical losses in the gas turbine are very small because of the few moving parts, nevertheless the main bearings and auxiliary drives such as oil pump, fuel pump, and governor drive claim a small amount of power. The net effect of the various losses is to reduce the over-all efficiency and output.

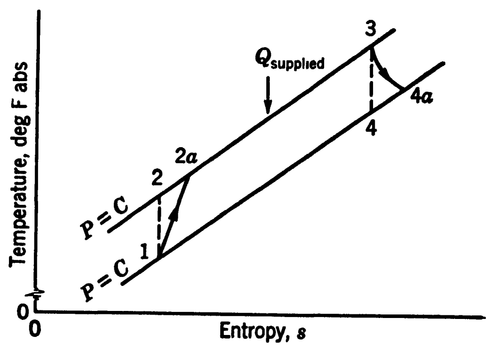


FIG. 247. Cycle with compressor and turbine losses.

270. Simple Cycle with Losses. A simple cycle in which only the machine losses (compressor and turbine) are assumed to occur is shown as solid lines in Fig. 247 in contrast to the ideal cycle shown as dotted lines.

The efficiency of this nonideal cycle is

$$e_a = \frac{W_{net}}{J \times Q_{supplied}} = \frac{W_t + W_c}{J \times Q_{supplied}} \qquad (298)$$

where e_a = cycle efficiency with compressor and turbine losses considered.

$$\frac{W_c}{J} = \text{compressor work} = (h_1 - h_{2,a}) = \frac{(h_1 - h_2)}{e_c} \quad \text{Btu per lb}$$

or

$$\frac{W_c}{J} = \frac{c_p(T_1 - T_2)}{e_c} \quad \text{Btu per lb} \qquad (299)$$

$$e_c = \text{compression efficiency} = \frac{W_{isentropic}}{W_{actual}} = \frac{h_1 - h_2}{h_1 - h_{2,a}}$$

$$e_c = \frac{T_1 - T_2}{T_1 - T_{2,a}} \tag{300}$$

$$\frac{W_t}{J} = \text{work of the turbine} = (h_3 - h_{4,a}) = (h_3 - h_4)(e_t) \quad \text{Btu per lb}$$

or

$$\frac{W_t}{J} = c_p(T_3 - T_4)(e_t) \quad \text{Btu per lb} \tag{301}$$

$$e_t = \text{turbine efficiency} = \frac{W_{actual}}{W_{isentropic}} = \frac{h_3 - h_{4,a}}{h_3 - h_4}$$

$$e_t = \frac{T_3 - T_{4,a}}{T_3 - T_4} \tag{302}$$

Substituting in the cycle-efficiency equation,

$$e_a = \frac{(h_3 - h_4)e_t + (h_1 - h_2)/e_c}{h_3 - h_{2,a}}$$

(Note that compressor work is negative.) Also, the efficiency of this simple cycle for a perfect gas with compressor and turbine losses considered is

$$e_a = \frac{e_t T_3 \left(1 - \frac{1}{r^{(k-1)/k}}\right) + \frac{T_1}{e_c}(1 - r^{(k-1)/k})}{T_3 - T_1\left(1 + \frac{r^{(k-1)/k} - 1}{e_c}\right)} \tag{303}$$

where $r = P_2/P_1$ and subscripts for P, T, and h refer to Fig. 247. Other terms are as defined previously.

The cycle air rate or specific air consumption depends on the turbine and compressor efficiencies and on the pressure ratio. The higher the pressure and the temperature, the less is the amount of air to be handled, and a smaller machine can be used. The *air rate* per horsepower-hour is the quotient of the Btu equivalent of a horsepower-hour and the net work per pound of air, $(W_t + W_c)/J$, or

$$\text{Air rate} = \frac{2545}{(W_t + W_c)/J} \quad \text{lb per hp-hr} \tag{304}$$

The *work ratio* is defined as the net work divided by the turbine

work; i.e.,

$$\text{Work ratio} = \frac{(W_t + W_c)}{W_t} \tag{305}$$

This ratio becomes smaller with high pressures and low machine efficiencies.

271. Regenerative Cycle with Losses. Regenerator effectiveness rarely exceeds 75 per cent because the heat-transfer surface needed becomes excessive and increases the resistance to flow of both air and gases. The power output, however, will not be improved. There are more occasions for losses in the regenerative cycle than in the simple cycle. The losses are due to the resistance to air and gas flow through the regenerator as well as to increased friction losses by virtue of the extra piping.

The effectiveness of a regenerator for a cycle involving losses is given by

$$e_e = \frac{T_{e,a} - T_{2,a}}{T_{4,a} - T_{2,a}} \tag{306}$$

where T = temperature, deg F abs; subscript a designates actual conditions; and other subscripts refer to state points in Figs. 245(b) and 247. If the effectiveness of a regenerator is known, equation 306 is useful in determining exit temperatures for the regenerator. Since thermal efficiency is the net output in Btu divided by the input in Btu

$$e_{ra} = \frac{(W_t + W_c)/J}{Q_{net}} = \frac{(W_t + W_c)/J}{Q_{nr} - Q_{rec}} \tag{307}$$

where e_{ra} = efficiency of regenerative cycle with turbine and compressor losses considered.

$\dfrac{W_c}{J}$ = work of compressor with losses, Btu per lb of air flow.

$\dfrac{W_t}{J}$ = work of turbine with losses, Btu per lb of air flow.

Q_{nr} = energy supplied for a nonregenerative cycle, Btu per lb of air flow.

Q_{rec} = energy recovered by regenerator, Btu per lb of air flow.

Example. Assuming a compressor and turbine efficiency of 83 per cent and 89 per cent, respectively, find the cycle efficiency (a) without regeneration and (b) with a regenerator having 75 per cent effectiveness. Intake pressure = 14.5 psia, intake temperature = 70 F, maximum temperature = 1500 F, mean specific heat for air and gases = 0.26, and pressure ratio = 4:1. Assume k = 1.4.

Solution. (*a*) Using equation 303,

$$e_a = \frac{0.89(1960)\left(1 - \frac{1}{4^{(1.4-1)/1.4}}\right) + \frac{530}{0.83}\left(1 - 4^{(1.4-1)/1.4}\right)}{1960 - 530\left[1 + \frac{4^{(1.4-1)/1.4} - 1}{0.83}\right]}$$

$$= 0.234, \quad \text{or} \quad 23.4 \text{ per cent}$$

(*b*) From $T_2/T_1 = T_3/T_4 = r^{(k-1)/k}$

$$T_2 = 787.6 \text{ F abs} \quad \text{and} \quad T_4 = 1319 \text{ F abs}$$

Since

$$e_c = \frac{T_2 - T_1}{T_{2,a} - T_1}$$

$$0.83 = \frac{787.6 - 530}{T_{2,a} - 530}$$

or

$$T_{2,a} = 841 \text{ F abs}$$

Since

$$e_t = \frac{T_3 - T_{4,a}}{T_3 - T_4} \quad \text{equation 302}$$

$$0.89 = \frac{1960 - T_{4,a}}{1960 - 1319}$$

or $$T_{4,a} = 1389.6 \text{ F abs}$$

Since

$$e_e = \frac{T_{e,a} - T_{2,a}}{T_{4,a} - T_{2,a}} \quad \text{equation 306}$$

$$0.75 = \frac{T_{e,a} - 841}{1389.6 - 841}$$

or

$$T_{e,a} = 1253 \text{ F abs}$$

Substituting in equation 301,

$$\frac{W_t}{J} = c_p(T_3 - T_4)(e_t)$$

$$\frac{W_t}{J} = 0.26(1960 - 1319)(0.89)$$

$$= 0.26(570.5) \text{ Btu per lb of air flow}$$

Substituting in equation 299,

$$\frac{W_c}{J} = \frac{c_p(T_1 - T_2)}{e_c}$$

$$\frac{W_c}{J} = \frac{0.26(530 - 788)}{0.83}$$

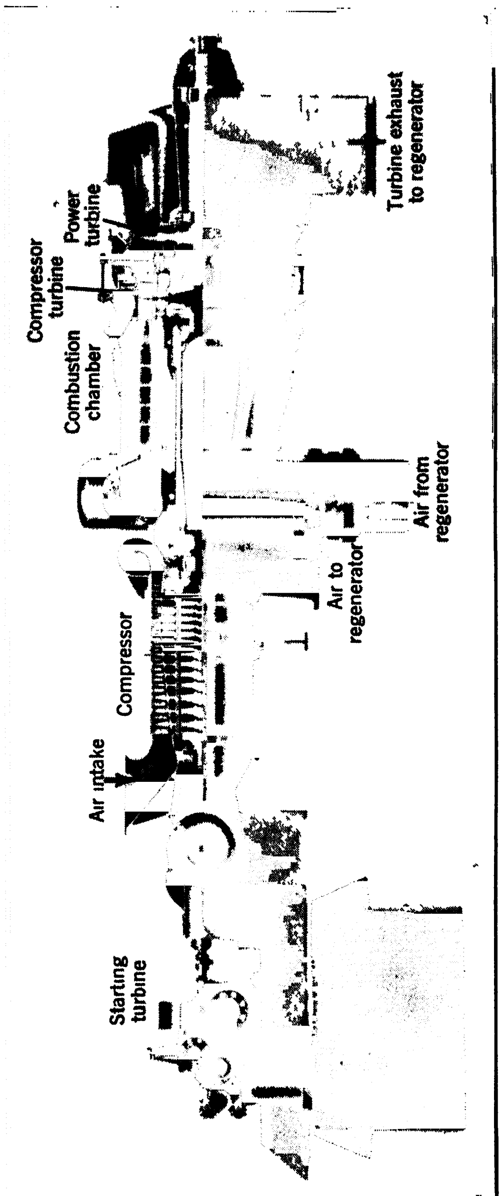


FIG. 248. Cross section of General Electric two-shaft 5000-hp regenerative-cycle turbine with independent load shaft for mechanical drive.

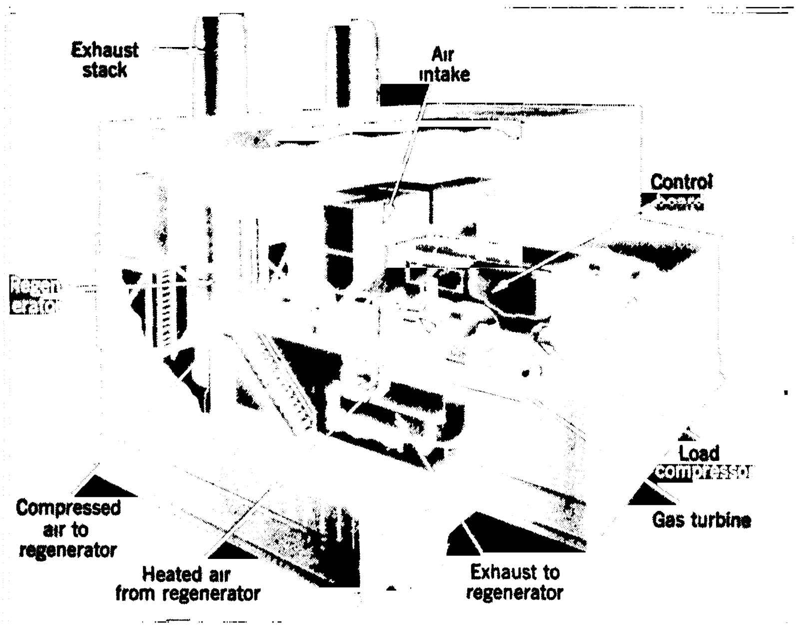


FIG. 249 General Electric self-contained 10,000-hp gas-turbine pumping station.

or

$$\frac{-W_c}{J} = 0.26(310\ 8) \text{ Btu per lb of air flow}$$

Since the energy supplied without regeneration is

$$mc_p(T_3 - T_{2,a})$$

and the energy recovered by regeneration is

$$mc_p(T_{e,a} - T_{2,a})$$

$$Q_{nr} = 0.26(1960 - 841) = 0\ 26(1119) \text{ Btu per lb of air flow}$$

$$Q_{rec} = 0.26(1253 - 841) = 0.26(412) \text{ Btu per lb of air flow}$$

Therefore, substituting in equation 307,

$$e_{ra} = \frac{0.26(570\ 5 - 310\ 8)}{0.26(1119 - 412)}$$

$$e_{ra} = 0.367, \quad \text{or} \quad 36.7 \text{ per cent}$$

Figure 248 shows a cross section of a regenerative-cycle gas turbine rated at 5000 hp. This unit, of the two-shaft design, has an axial-flow compressor driven by the low-pressure turbine. Figure 249 shows two of these units as they might be installed in a compact gas pumping plant with the regenerators inside the stacks. For steam generation

the unit is modified further by a two-stage compressor with an intercooler. A flow diagram and heat balance for an all-gas-turbine power plant (the Edward M. Graham Station, Bangor, Me.) is shown in Fig. 250. The unit handles 72 lb of air per sec, and the maximum temperature is 1500 F. During operation, No. 6 fuel oil is used but a lighter oil is used for starting. The fuel is burned in six combustion chambers with discharge directly into the high-pressure turbine nozzles.

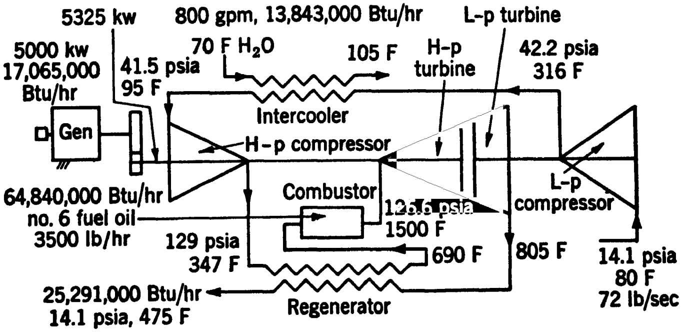


FIG. 250. Cycle and partial heat-balance diagram for 5000-kw gas turbine at 80 F and 1000 ft above sea level.

The high-pressure turbine operates at 8700 rpm and drives the generator at 3600 rpm. The low-pressure turbine operates normally at 7700 rpm, and since it drives only the low-pressure compressor its speed is allowed to vary with the load. Two regenerators are used, each having 2500 tubes, 20 ft in length and 1 in. in diameter with a total surface of 26,200 sq ft. The intercoolers provide counterflow with an effectiveness of 90 per cent.[2]

272. Ericsson Cycle. With sufficient stages of intercooling in the compressor, isothermal compression is approached; likewise, with a sufficient number of reheat stages in the turbine, isothermal expansion is approached. A large number of intercooler and reheat stages are impractical, particularly in a small unit; however, in large stationary installations several stages of compression with intercooling and at least two stages of expansion with one reheat are common practice.

Assuming that isothermal compression and isothermal expansion could be approached as indicated in Fig. 251 and that perfect regeneration could be obtained, i.e., that the energy from the exhaust, 4 to 1, would be adequate to heat the compressed air from 2 to 3, then the only energy to be added would occur from 3 to 4, and the only energy discarded would be that from 1 to 2.

[2] *Power Engineering*, March, 1951, pp. 58–60.

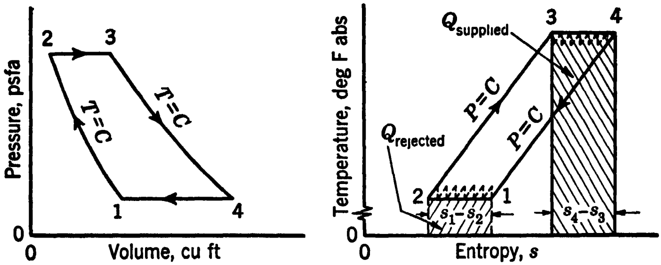


FIG. 251. Ericsson cycle.

The ideal cycle efficiency would then be

$$e = \frac{Q_{supplied} - Q_{rejected}}{Q_{supplied}}$$

$$e = \frac{T_3(s_4 - s_3) - T_1(s_1 - s_2)}{T_3(s_4 - s_3)} = \frac{T_3 - T_1}{T_3} \tag{308}$$

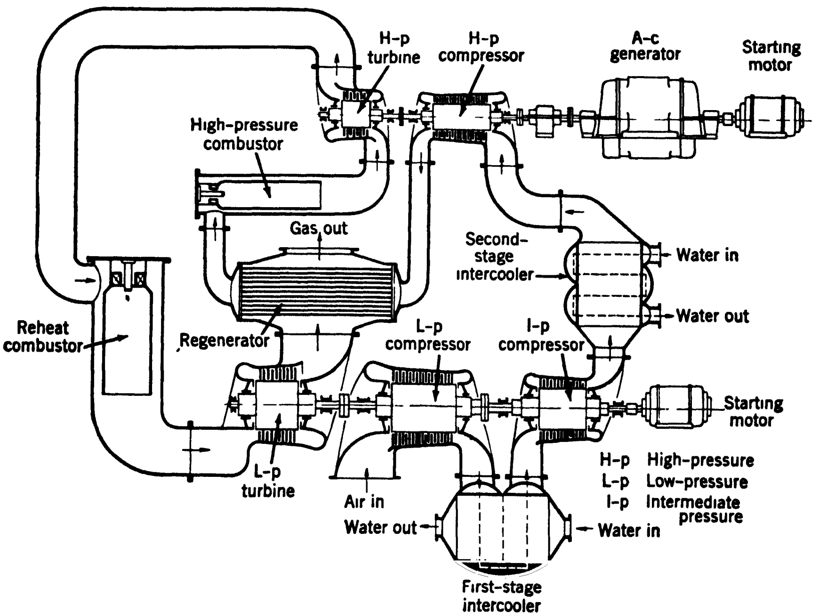


FIG. 252. Brown Boveri two-shaft open-cycle gas turbine.

where s = entropy, Btu per (lb) (deg F). Thus, the Ericsson cycle gives the same efficiency as the Carnot cycle since all the energy discarded is at minimum temperature.

273. Cycle Modifications. Many modifications may be incorporated in a gas-turbine power plant such as regeneration, Art. 267, multistage compression with intercooling, and multistage expansion with reheat.

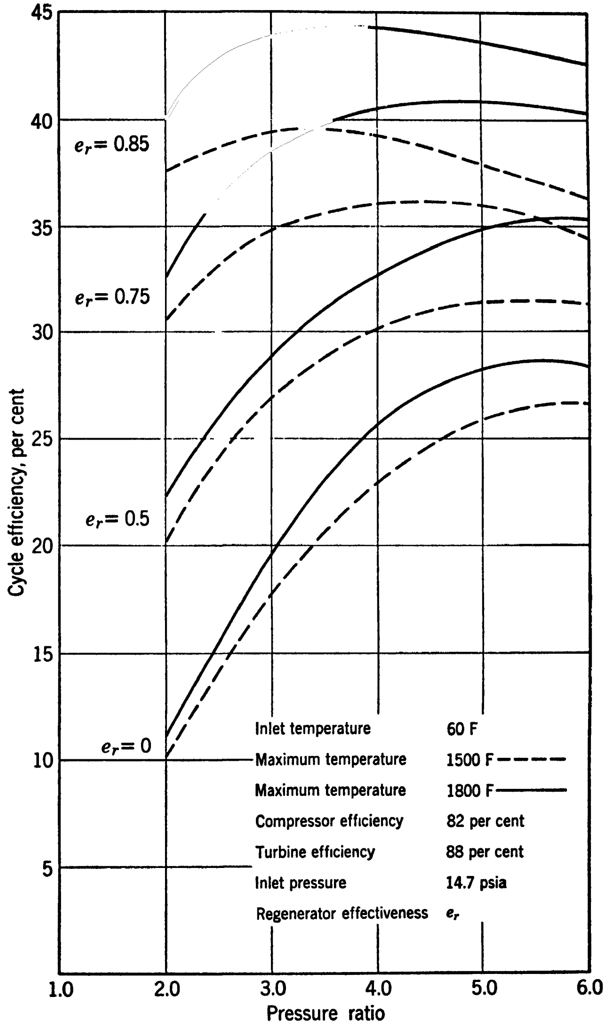


FIG. 253. Effect of pressure ratio, maximum cycle temperature, and regenerator effectiveness on cycle efficiency.

Figure 252 shows a two-shaft unit in which intercoolers are used between the three compressors; the air then passes through the tubular-type heat exchanger before entering the first-stage combustor, the first-

stage turbine, the second-stage combustor, the second-stage turbine, and again through the regenerator. The first-stage turbine drives the high-pressure compressor and the alternator. The low-pressure turbine drives the first- and second-stage compressors. Each of the two shafts is equipped with an electrical starting motor. The advantages of stage compression with intercooling have been discussed under Art. 255. With large volumes of air flow, multistage compression provides an effective means of decreasing the work of compression, especially if adequate quantities of cooling water are available.

Reheating the gases after partial expansion in the turbine increases the enthalpy of the gases before final expansion, and the temperature limits of the materials of construction are not exceeded. The combination of regeneration, reheat, and intercooling allows actual thermal efficiencies (based on the higher heating value of the fuel oil) of the order of 33 per cent.

Figure 253 shows the effect of temperature, pressure ratio, and regenerator effectiveness on the over-all cycle efficiency for typical values of compressor and turbine efficiencies. Increased regenerator effectiveness produces a rapid increase in cycle efficiency for low pressure ratios. Above the pressure ratio 4:1 the efficiency of the simple cycle approaches that of the regenerative cycle and exceeds it at the ratio of approximately 12:1. Above this ratio the regenerator is of zero value.

PROBLEMS

1. Determine the ideal air-standard cycle efficiency for a simple gas turbine when operating at the following pressure ratios: 3:1, 6:1, 9:1, 12:1. Plot efficiency vs. pressure ratio.

2. A simple gas turbine with inlet air at 14.7 psia and 60 F has a compression ratio of 3.5:1 and a maximum temperature of 1500 F. Assuming isentropic compression and expansion and an air flow of 20 lb per sec, determine (*a*) the power developed by the turbine, (*b*) the power required by the compressor, (*c*) the Btu supplied per second to heat the air, and (*d*) the cycle efficiency. Assume constant mean specific heat for both air and gases of 0.25. $k = 1.4$.

3. A simple gas turbine has a pressure ratio of 4:1 and uses 60 lb of air per sec. Intake pressure = 14.7 psia, intake temperature = 60 F, maximum temperature = 1600 F. Find (*a*) the cycle efficiency, (*b*) the net work done per second, (*c*) the theoretical horsepower, and (*d*) the work ratio. Assume ideal air-standard conditions.

(*a*) 32.75 per cent, (*b*) 6084 Btu per sec, (*c*) 8610 hp, (*d*) 0.63, *Ans.*

4. Using the operating conditions of problem 3 but assuming compressor and turbine efficiencies of 85 per cent each, find (*a*) the cycle efficiency, (*b*) the net work done per second, (*c*) the theoretical horsepower, and (*d*) the work ratio.

5. Sketch the *Ts* diagram to scale for the cycle in problem 4. Show energy quantities to scale.

6. An ideal regenerative-type gas turbine has a pressure ratio of 4:1 and uses 60 lb of air per sec. Intake pressure = 14.7 psia, intake temperature = 60 F,

maximum temperature = 1600 F. Find (*a*) the cycle efficiency, (*b*) the net work done per second, (*c*) the theoretical horsepower, and (*d*) the work ratio. Assume constant mean specific heat for both air and gases of 0.25. $k = 1.4$.

7. Sketch the Ts diagram to scale for the cycle in problem 6, showing energy supplied and rejected as well as that transferred by the regenerator.

8. For a pressure ratio of 5:1 with $T_1 = 500$ F abs and $T_3 = 1600$ F abs the remaining two temperatures of a simple ideal gas-turbine cycle are $T_2 = 793$ F abs and $T_4 = 1010$ F abs. For an air flow of 20 lb per sec, find the work and efficiency without and with regeneration.

✓ **9.** A turbine uses 20 lb of air per sec at a pressure ratio of 4:1. Assuming a regenerator effectiveness of 70 per cent, turbine and compressor efficiencies of 87 per cent each, constant mean specific heat of 0.25 for both air and gases, and neglecting mechanical losses, find (*a*) temperatures of air entering and leaving regenerator; (*b*) temperatures of gases entering and leaving the regenerator, and (*c*) the mean temperature difference. Inlet pressure = 14.7 psia, inlet temperature = 60 F, and temperature entering turbine = 1600 F.

(*a*) 811 deg F abs, 1275 deg F abs; (*b*) 1474 deg F abs, 1010 deg F abs; (*c*) 199 F; *Ans.*

10. A simple gas turbine takes in air at 14.7 psia and 60 F and compresses it through a pressure ratio of 5:1. The compression efficiency is 85 per cent. In the combustion chamber the air is heated to 1500 F and then expanded to 14.7 psia through a turbine having an efficiency of 90 per cent Determine the air rate in pounds per second necessary to produce 1000 shp, assuming no loss in pressure through the combustion chamber and neglecting the mass of the fuel in the air stream. The value of c_p for both air and gases is 0.25. $k = 1.4$.

11. A gas turbine compresses air at 60 F and 14.7 psia, the pressure ratio being 3.5:1 and the maximum temperature 1600 F. Assuming efficiencies of 82 and 88 per cent, respectively, for the compressor and the turbine, determine the efficiency (*a*) without regeneration and (*b*) with a regeneration effectiveness of 80 per cent. Neglect pressure and mechanical losses in the system, and assume the medium to be pure air. Assume $c_p = 0.25$ for both air and gases.

12. Determine the square feet of heat-exchanger area required for a counterflow tubular-type heat exchanger having a regenerator effectiveness of 80 per cent when receiving exhaust gases at 1200 F and air from the compressor at 380 F. Assume the weight of air and gas flow to be the same, 72 lb per sec, the over-all coefficient of heat transfer to be 5.0 Btu per (hr) (sq ft) (deg F), and $c_p = 0.25$ for both gas and air.

51,800 sq ft, *Ans.*

13. Using the data in problem 12, determine the number of ½-in. O.D. tubes 22 ft long required to provide the necessary area, based on the exterior surface of the tubes.

CHAPTER 16

Internal-Combustion Engines

274. Development. A reciprocating internal-combustion (IC) engine consists primarily of a cylinder, a piston, and a crank. Fuel is burned within the cylinder, and work is done by the expanding gaseous products of combustion. Rotary motion is obtained by means of the crank. The earliest known use of a cylinder and piston is ascribed to the Malayan aborigines who produced fire by quick compression of a plunger in a bamboo tube. Later the concept of the wheel became known, and the Dutch physicist Huyghens is credited with an attempt to generate power by burning gunpowder in a cylinder in 1680. About two hundred years later Lenoir, a Frenchman, developed the first workable, but inefficient, gas engine. In 1862 another Frenchman, Beau de Rochas, patented a cycle which was incorporated into a four-stroke-cycle engine by Otto and Langen of Germany. This engine was the first of the Otto-cycle engines. The two-stroke-cycle principle was conceived by Dugald Clerk of England in 1879. In 1872 George Brayton patented and built an engine in which combustion occurred at constant pressure. Later the Brayton cycle found application in the gas turbine. Rudolf Diesel in 1892 conceived the idea of igniting the fuel by compression rather than by spark or open flame.

275. Nomenclature. There are many auxiliary systems necessary to the proper functioning of an engine. Table XI shows some of these systems and their variations. The parts of the internal-combustion engine have acquired names from usage. Figure 254 shows a sectional view of a six-cylinder four-stroke-cycle Diesel engine. Figure 255 shows a sectional view of a Diesel tractor, 180 hp for intermittent service and 144 hp for continuous service. The engine has 1090.6-cu-in. displacement and is rated at 1375 rpm.

276. Mechanical Cycles. A cycle is a series of operations in which the apparatus or substances periodically return to one given state or configuration. A mechanical cycle for an internal-combustion engine can be completed in one revolution (two-stroke cycle) or in two revolutions (four-stroke cycle). In Otto-cycle engines the fuel and air are

TABLE XI

Internal-Combustion Engine System Variations

- **Cylinder and piston**
  - 1. Cycle
    - Otto
    - Diesel
    - Lenoir
    - Brayton
  - 2. Type
    - Single-acting
    - Double-acting
    - Opposed-piston
      - Single-crank
      - Double-crank
    - Trunk-piston
    - Skirt-piston
    - Two-stroke-cycle
    - Four-stroke-cycle
  - 3. Cylinder arrangement and number
    - In-line 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12
    - V-type 2, 4, 8, 12, 16
    - Radial 2, 3, 5, 7, 9, 11, 12
    - Radial in Banks of 1, 2, 3, 4
    - X-type
    - Barrel type
  - 4. Cylinder position
    - Horizontal
    - Vertical
    - Inverted
  - 5. Valves
    - Poppet
      - Overhead
      - L-head
      - T-head
      - F-head
    - Sleeve
    - Rotary
    - Ports
  - 6. Ignition
    - Spark
      - Magneto
      - Battery
    - Compression
    - Glow plug
  - 7. Starting
    - Battery
    - Compressed air
    - Inertia
    - Cartridge
    - Motoring
  - 8. Cooling
    - Liquid
      - Wet liner
      - Dry liner
    - Air
  - 9. Breathing
    - Naturally aspirated
    - Supercharged
      - Gear driven
      - Exhaust driven
    - Scavenged
      - Cross, Loop, Through
        - Blower
        - Piston type
        - Crank case
  - 10. Fuel System
    - Carburetion
      - Gasoline
      - LP gas
      - Natural gas
    - Fuel injection
      - Oil
        - Solid
        - Air
      - Gasoline
        - Manifold
        - Direct
    - Combination (oil and gas)
  - 11. Lubrication
    - Wet sump
    - Dry sump
    - Pressurized

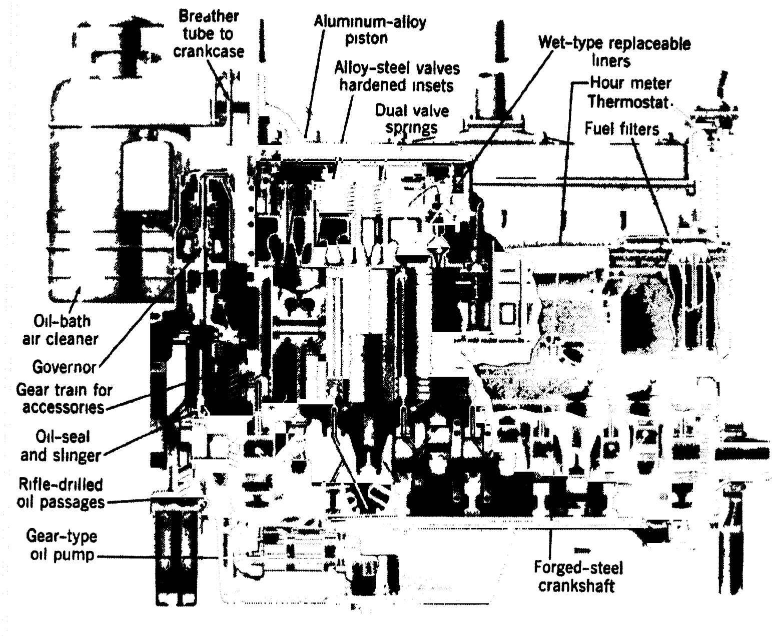


FIG 254 Caterpillar D337 Diesel engine

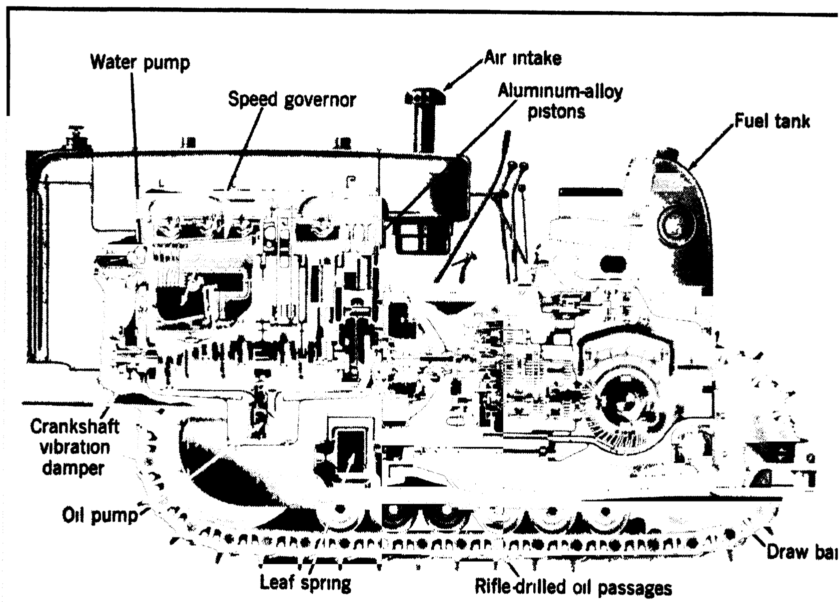


FIG 255 International Harvester TD24 tractor.

mixed in a carburetor and ignited by a spark; in Diesel-cycle engines the fuel is injected by a pump or by compressed air and ignited by the heat of compression. Neither fully complies with the thermodynamic

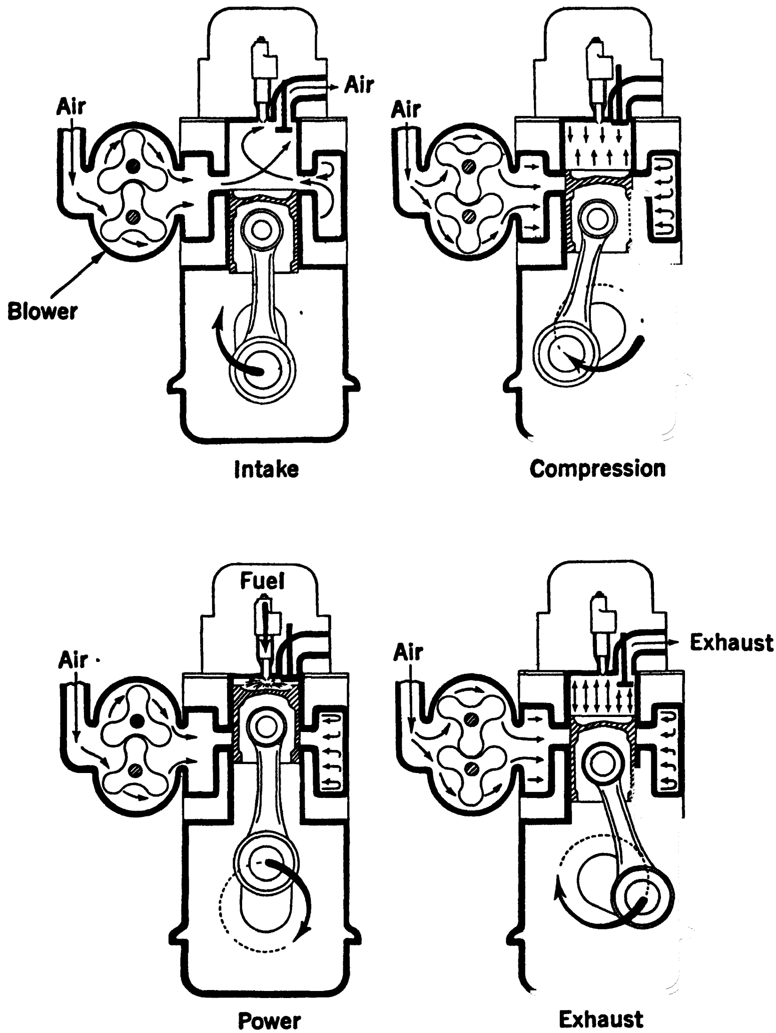


FIG. 256. Illustration of two-stroke Diesel cycle.

cycle for which it was named. Actually, an internal-combustion engine does not operate on a thermodynamic cycle because the medium does not return to the original state point but is discarded through the exhaust, and a new charge is brought into the cylinder. For thermody-

namic analysis a closed cycle is assumed in which the same medium is repeatedly carried through the cycle.

TWO-STROKE CYCLE. Figure 256 illustrates a two-stroke-cycle Diesel engine in which a blower is used to scavenge the cylinder and supply a

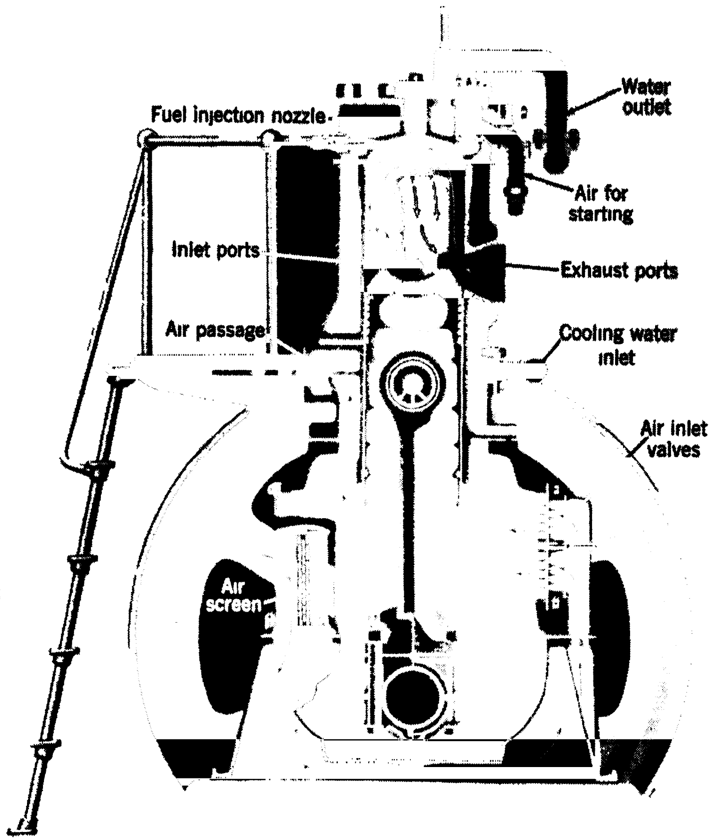


FIG 257 Fairbanks-Morse crankcase-scavenged engine.

fresh charge of air. The blower is driven from the engine shaft. Air enters at the ports under a pressure of approximately 4 psig, and the exhaust gases are driven out at the top of the cylinder through poppet-type exhaust valves. The engine generates a complete cycle in one revolution. Diesel engines are well adapted to two-stroke-cycle operation because the in-going air carries no fuel and any air escaping with the exhaust gases (blow-through) represents only the loss of work done

on the air. If the escaping air carries gasoline vapor, as in a carburetted (Otto) engine, any loss of air also causes a serious loss of fuel.

Small gasoline engines, such as outboard motors, are frequently of the two-stroke-cycle type. In this size range efficiency is relatively insignificant. By using an exhaust port and crankcase compression, the need for camshaft, timing gears, poppet valve, push rod, and rocker arm is eliminated.

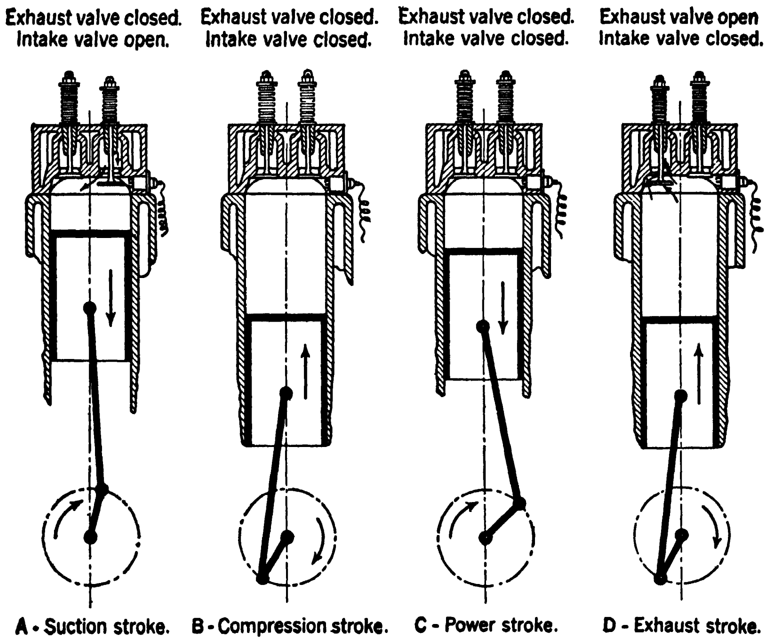


FIG. 258. Otto four-stroke-cycle engine.

More Diesel engines of the two-stroke-cycle type have been built than of the four-stroke-cycle type. The primary advantages claimed are (1) less frictional loss per horsepower, (2) compactness, and (3) simpler construction. The claim of simpler construction may be challenged since a scavenging blower is required unless crankcase scavenging is employed. The latter necessitates a bulkier engine per horsepower, because the limited air supply in the crankcase produces ineffective scavenging and the cylinder size needs to be doubled for equivalent power output. Figure 257 shows a two-stroke-cycle engine of the crankcase-scavenged type.

FOUR-STROKE CYCLE. Figure 258 illustrates the four-stroke-cycle Otto engine in which the mechanical cycle is completed in four strokes

of the piston or two revolutions of the crankshaft. In *A* is shown the suction stroke in which a charge of air and fuel is drawn into the cylinder. In *B* the mixture is being compressed with both intake and exhaust valves closed. When the piston nears the top of its travel (head-end dead center), the ignition system provides a spark and causes combustion of the compressed mixture. The resulting high pressure from the heated gases forces the piston down on the power stroke *C*. Near the end of the power stroke (crank-end dead center) the exhaust valve opens and allows the spent gases to escape. The piston rises again on the exhaust stroke *D*, forcing the rest of the exhaust gases out and completing the cycle.

The four-stroke-cycle Diesel engine operates on the same principle except that the fuel is forced into the cylinder near the end of the compression stroke. The fuel is ignited by the air that has been heated by compression.

277. Thermodynamic Cycles. All engines have had their origin in some concept of a thermodynamic cycle. This is a series of events in which energy is received at a high level, as much as possible is converted to work, and the rest is dumped to the surroundings (second law of thermodynamics).

For the purpose of analysis, it is customary to consider air as the working medium, also to assume that it is an ideal medium, i.e., that its specific heat is constant and it obeys exactly the laws of a perfect gas. A cycle based on such assumptions is called *air-standard*, or *ideal*, or *theoretical*, or any combination of these names.

Other assumptions are usually made to simplify the analysis. For instance, all the cycle events are assumed to be reversible, thereby eliminating all frictional effects. Also, any process may be designated as adiabatic or isothermal. In practice these things cannot be realized. The efficiency of a thermodynamic cycle when figured on the above basis will be roughly twice as great as that obtained from a real engine cycle. The air-standard efficiency serves as a first approximation and points the way to changes by which the cycle may be improved. Of the various thermodynamic cycles the Carnot cycle, Art. 48, serves as the standard of perfection. Although it cannot be applied to a real engine it is extremely valuable in understanding the factors by which efficiency and work are affected.

278. Air-Standard Otto Cycle. The conditions for the Otto cycle, as conceived by Beau de Rochas, were:

1. Largest cylinder volume with smallest exposed surface to limit heat transfer.

2. Highest allowable pressure at the start of the expansion or power stroke.
3. Maximum piston speed to further limit heat transfer.
4. Greatest possible expansion.

As stated earlier, only a mechanical cycle is executed in a real engine since the gases are exhausted to the surroundings. Assuming a thermodynamic cycle, a closed cylinder can be envisioned in which the air is heated and cooled by energy transfer rather than by combustion and exchange of air. Such a cycle is illustrated in Fig. 259 in which the

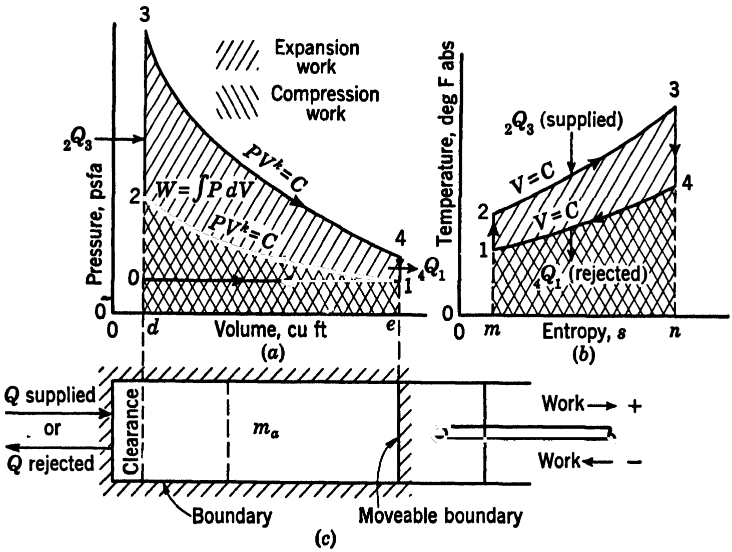


FIG. 259 Air-standard Otto cycle

work done appears on the PV plane, and the energy supplied and rejected appears on the Ts plane.

In Fig. 259, assume that a quantity of air (m_a) is contained in the cylinder at all times; therefore the suction and exhaust strokes (0 to 1 and 1 to 0) need not be considered. The ideal cycle assumes isentropic compression (1 to 2), constant-volume energy addition (2 to 3), isentropic expansion (3 to 4), and constant-volume energy rejection (4 to 1). All processes are reversible and the air acts as a perfect gas with specific heats constant.

The air-standard Otto-cycle efficiency is given by

$$e_o = \frac{{}_2Q_3 - {}_4Q_1}{{}_2Q_3} = \frac{W/J}{{}_2Q_3} \tag{309}$$

where ${}_2Q_3$ = energy supplied = $m_a c_v(T_3 - T_2)$, Btu.
${}_4Q_1$ = energy rejected = $m_a c_v(T_4 - T_1)$, Btu.
$\frac{W}{J}$ = net work per cycle = ${}_2Q_3 - {}_4Q_1$, Btu.

$$e_o = \frac{m_a c_v(T_3 - T_2) - m_a c_v(T_4 - T_1)}{m_a c_v(T_3 - T_2)}$$

or

$$e_o = 1 - \frac{T_4 - T_1}{T_3 - T_2}$$

Factoring out T_1/T_2,

$$e_o = 1 - \frac{T_1\left(\frac{T_4}{T_1} - 1\right)}{T_2\left(\frac{T_3}{T_2} - 1\right)}$$

From the laws of perfect gases, the temperature-volume relations for the isentropic processes 1 to 2 and 3 to 4, Fig. 259, are

$$\frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1}$$

and

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1}$$

Since $V_3 = V_2$ and $V_4 = V_1$, it follows that

$$\frac{T_1}{T_2} = \frac{T_4}{T_3} \quad \text{or} \quad \frac{T_4}{T_1} = \frac{T_3}{T_2}$$

and

$$e_o = 1 - \frac{T_1}{T_2} = 1 - \left(\frac{V_2}{V_1}\right)^{k-1} = 1 - \frac{1}{r^{k-1}} \qquad (310)$$

where $r = V_1/V_2$, the *compression ratio.*

$k = c_p/c_v = 1.4$ for air.

A consideration of equation 310 shows that e_o becomes larger as T_2 becomes larger with respect to T_1, also as V_1 becomes larger with respect to V_2; i.e., as the compression ratio becomes greater. In practice, the maximum-compression ratio that is built into an Otto engine is dictated by the grade of fuel that will be used in it, Art. 84. Consequently, its efficiency is limited by the maximum compression ratio that the fuel will withstand. It is interesting to note also that as k becomes larger the cycle efficiency is greatly improved.

From Fig. 259(*b*), the total energy supplied per cycle is *m*–2–3–*n*–*m* Btu, the energy rejected per cycle is *m*–1–4–*n*–*m* Btu, and the net usable energy per cycle is 1–2–3–4–1 Btu. From Fig. 259(*a*), the work of compression is 1–2–*d*–*e*–1 ft-lb per cycle, the work of expansion is 4–3–*d*–*e*–4 ft-lb per cycle, and the net work per cycle is 1–2–3–4–1 ft-lb per cycle.

The net work may also be obtained for the complete cycle by the integration of $P\,dV$ around the cycle,

or

$$W = \int_1^2 P\,dV + \int_2^3 P\,dV + \int_3^4 P\,dV + \int_4^1 P\,dV \quad \text{ft-lb}$$

$$W = \frac{P_2V_2 - P_1V_1}{1 - k} + 0 + \frac{P_4V_4 - P_3V_3}{1 - k} + 0$$

$$\frac{W}{J} = \frac{-m_aR}{(k-1)J}[(T_2 - T_1) + (T_4 - T_3)]$$

$$\frac{W}{J} = m_ac_v[(T_3 - T_2) - (T_4 - T_1)] \quad \text{Btu}$$

$$\frac{W}{J} = {}_2Q_3 - {}_4Q_1 \quad \text{Btu}$$

Therefore

$$e_o = \frac{W/J}{{}_2Q_3} \tag{311}$$

279. Air-Standard Diesel Cycle. This cycle, Fig. 260, assumes isentropic compression (1 to 2), energy addition at constant pressure (2 to 3), isentropic expansion (3 to 4), and constant-volume energy rejection (4 to 1). It is also assumed that air is the only medium in the process and that it is a perfect gas, $k = 1.4$. All operations are assumed to be reversible.

As shown in Fig. 260, the cycle efficiency is given by

$$e_d = \frac{{}_2Q_3 - {}_4Q_1}{{}_2Q_3} = \frac{W/J}{{}_2Q_3} \tag{312}$$

where ${}_2Q_3$ = energy supplied = $m_ac_p(T_3 - T_2)$, Btu.

${}_4Q_1$ = energy rejected = $m_ac_v(T_4 - T_1)$, Btu.

$\frac{W}{J}$ = net work per cycle = ${}_2Q_3 - {}_4Q_1$, Btu.

$$e_d = \frac{m_ac_p(T_3 - T_2) - m_ac_v(T_4 - T_1)}{m_ac_p(T_3 - T_2)}$$

or

$$e_d = 1 - \frac{c_v}{c_p}\left(\frac{T_4 - T_1}{T_3 - T_2}\right) = 1 - \frac{1}{k}\left(\frac{T_4 - T_1}{T_3 - T_2}\right)$$

For convenience this equation can be expressed in terms of the volumes. Factoring out T_1/T_2,

$$e_d = 1 - \frac{T_1}{T_2}\left\{\frac{\frac{T_4}{T_1} - 1}{k\left(\frac{T_3}{T_2} - 1\right)}\right\}$$

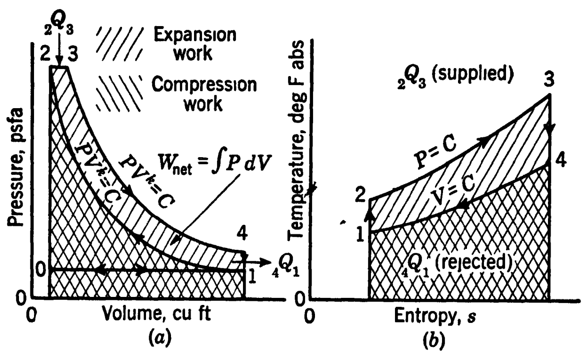


FIG 260 Air-standard Diesel cycle.

From the laws of perfect gases,

$$\frac{P_1 V_1}{T_1} = \frac{P_2 V_2}{T_2} = \frac{P_3 V_3}{T_3} = \frac{P_4 V_4}{T_4} \quad \text{for a given weight of gas}$$

or

$$\frac{T_4}{T_1} = \frac{P_4 V_4}{P_1 V_1} \quad \text{and} \quad \frac{T_3}{T_2} = \frac{P_3 V_3}{P_2 V_2}$$

Since $V_4 = V_1$ and $P_3 = P_2$, it follows that

$$\frac{T_4}{T_1} = \frac{P_4}{P_1} \quad \text{and} \quad \frac{T_3}{T_2} = \frac{V_3}{V_2}$$

For the isentropic processes 1 to 2 and 3 to 4,

$$P_4 V_4^k = P_3 V_3^k \quad \text{or} \quad P_4 = P_3\left(\frac{V_3}{V_4}\right)^k$$

and

$$P_1 V_1^k = P_2 V_2^k \quad \text{or} \quad P_1 = P_2 \left(\frac{V_2}{V_1}\right)^k$$

or

$$\frac{P_4}{P_1} = \left(\frac{V_3}{V_2}\right)^k \quad \text{also} \quad \frac{T_1}{T_2} = \left(\frac{V_2}{V_1}\right)^{k-1}$$

Therefore

$$e_d = 1 - \left(\frac{V_2}{V_1}\right)^{k-1} \left[\frac{\left(\frac{V_3}{V_2}\right)^k - 1}{k\left(\frac{V_3}{V_2} - 1\right)}\right] \tag{313}$$

or

$$e_d = 1 - \frac{1}{r^{k-1}} \left[\frac{\left(\frac{V_3}{V_2}\right)^k - 1}{k\left(\frac{V_3}{V_2} - 1\right)}\right]$$

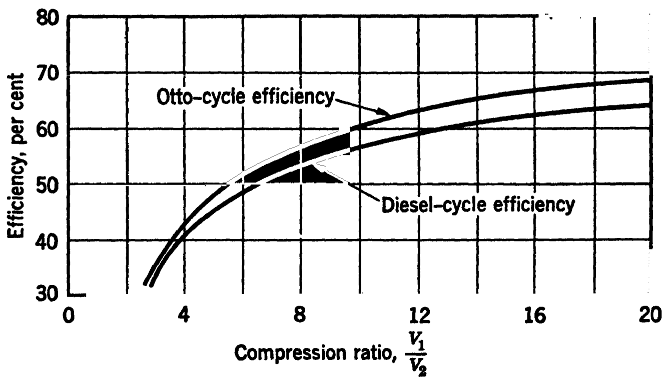


FIG. 261. Air-standard cycle efficiencies.

For the cycle efficiency to be maximum it is evident that the compression ratio r should be maximum. Increasing the compression ratio beyond 16:1 provides little gain in efficiency but results in very high cylinder pressures with consequent mechanical stresses on piston ring lands and on bearings. Most Diesel engines use compression ratios between 12 and 16 to 1. Figure 261 shows the relation between compression ratio and air-standard efficiency for the Otto and Diesel cycles. It is evident that for the same compression ratio, the Otto cycle is more

efficient because of the constant-volume burning assumed in the Otto. The Diesel cycle, however, is not limited to the same compression ratio as the Otto. High compression assists the burning operation in a Diesel engine and hinders burning or produces detonation in an Otto engine. Actual thermal efficiencies are higher in the Diesel than in the Otto, typical values being 30 to 35 per cent for the former and 20 to 27 per cent for the latter.

Further examination of equation 313 shows that the efficiency would be improved if k were larger. This possibility is beyond control as long as the oxygen is taken from the air. The value of k for nitrogen and other diatomic gases is essentially 1.4 at standard conditions and decreases to approximately 1.3 at temperatures encountered during combustion. The inert nitrogen brought in with the oxygen serves no useful purpose except to reduce the temperature. It absorbs energy released by the combustion of fuel and oxygen; then acting as a "reasonably" perfect gas it expands against the piston, thereby doing some work. The nitrogen is thus only a diluting agent, and any other inert gas having a higher value of k, such as helium ($k = 1.67$), would be better. Monatomic gases would not dissociate at the high temperatures encountered in a combustion chamber, as does nitrogen. The specific heat of monatomic gases is essentially constant.

The air-standard Diesel efficiency, Fig. 260, also improves as V_3/V_2, called the *load ratio*, decreases. V_2 is the *clearance volume* and V_3 is the *cut-off volume*. Therefore, at lighter loads the cycle efficiency is higher because the expansion ratio is greater; however, V_3 cannot be reduced very much before the mechanical efficiency decreases to a point where the over-all thermal efficiency begins to decrease. The lowest fuel consumption usually occurs near full load conditions. The shaft output is greatest in comparison to the friction losses under such conditions.

Example. Determine the air-standard efficiency of an engine operating on the Diesel cycle when the suction pressure is 14.5 psia and the fuel is injected for 5 per cent of the stroke. The clearance volume is 8 per cent of the stroke. Assume $k = 1.4$.

Solution. From equation 313

$$e_d = 1 - \left(\frac{0.08}{1.08}\right)^{0.4}\left[\frac{\left(\frac{0.13}{0.08}\right)^{1.4} - 1}{1.4\left(\frac{0.13}{0.08} - 1\right)}\right]$$

$$e_d = 0.607, \quad \text{or} \quad 60.7 \text{ per cent}$$

Example. Same as above, but assume $k = 1.67$.

Solution.

$$e_d = 1 - \left(\frac{0.08}{1.08}\right)^{0.67}\left[\frac{\left(\frac{0.13}{0.08}\right)^{1.67} - 1}{1.67\left(\frac{0.13}{0.08} - 1\right)}\right]$$

$$e_d = 0.791, \quad \text{or} \quad 79.1 \text{ per cent}$$

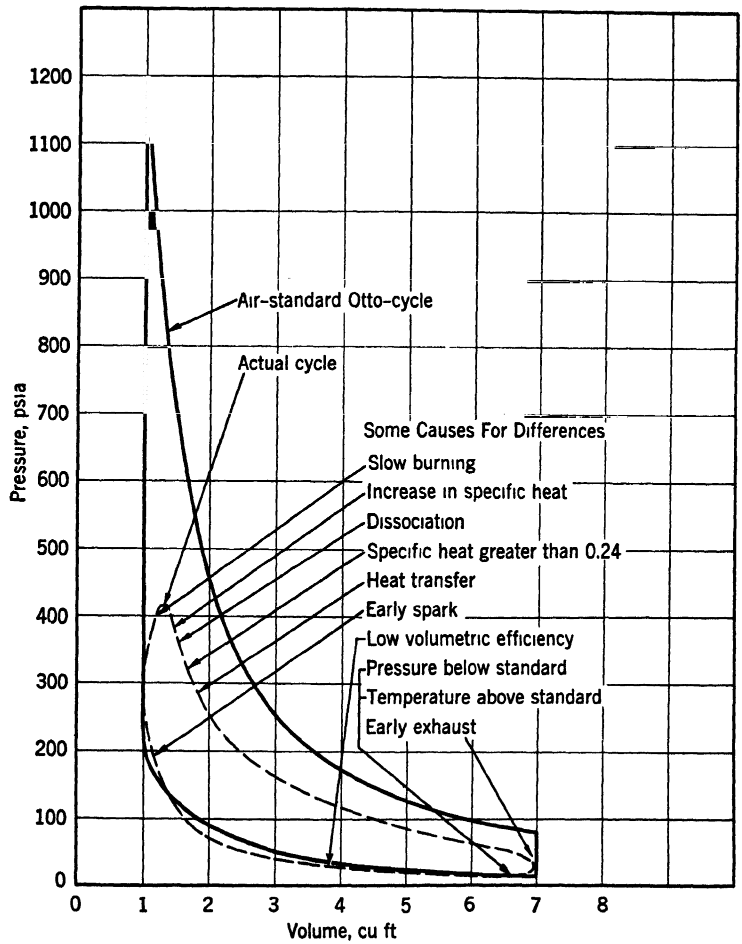


FIG. 262 Comparison of actual and ideal Otto cycles.

280. Deviations from the Air-Standard Cycle. Many factors cause the actual work to be less than that assumed in the air-standard cycle. Figure 262 shows an air-standard Otto cycle on which a typical indi-

cator diagram has been superposed. Some of the major causes for the differences between them are indicated. The importance of a small deviation in the cycle should not be minimized because in operation it is repeated in six or eight cylinders and two or three thousand times per minute. Thus a small deviation from the optimum setting in the

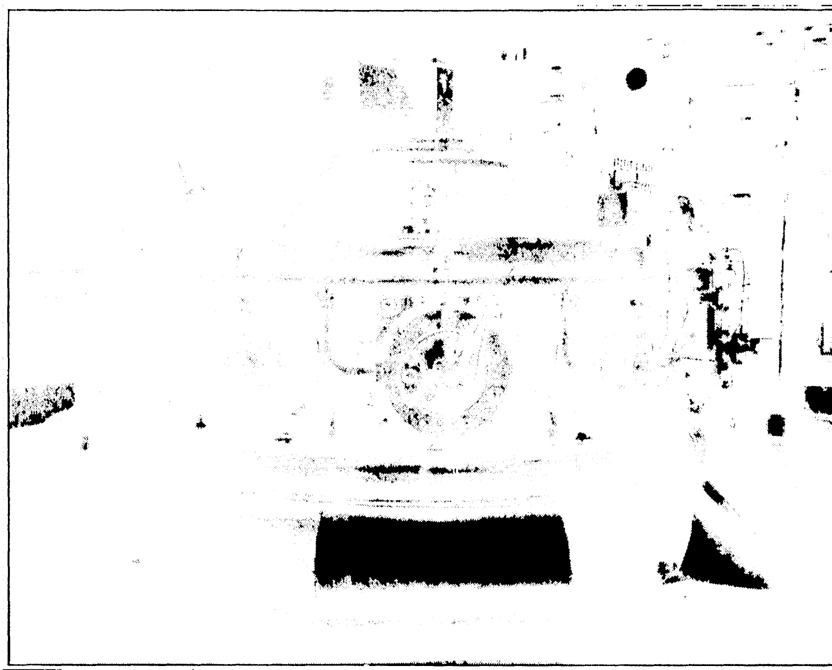


FIG 263 Nordberg 12-cylinder radial engine.

spark timing will cause a loss in efficiency and power that is significant. It should be recalled that an accurate PV diagram is a picture of the work generated in the cylinder. Any increase in engine work output must appear as an increase in area on the PV diagram.

The air-standard Diesel cycle is subject to most of the causes for deviation as pictured for the Otto cycle.

281. Dual-Fuel Engines. Figure 263 shows a two-stroke-cycle radial engine of 14-in. bore and 16-in. stroke. This engine delivers 2125 hp as a 12-cylinder unit operating at 400 rpm. It is designed to be easily converted from gas to oil or to a combination of oil and gas. As a gas engine, it employs spark ignition. As an oil-Diesel engine, compression ignition is used. As a gas-Diesel engine, compression ignition is used with the oil serving as the "pilot" fuel. The gas is mixed

and compressed with the air as a lean mixture through most of the compression stroke. The lean air-gas mixture will not spontaneously ignite at the compression pressure and the injection of a small amount of oil is necessary to start combustion. The oil is injected so that burning occurs essentially at head-end dead center.

The gas-Diesel engine offers several unique advantages.

1. Thorough mixing of the air and gas before combustion.
2. Clean burning with utilization of all the air when needed.
3. Smooth combustion with little tendency to knock.
4. Less tendency to contaminate or foul the lubricating oil.
5. The high efficiency of the Diesel cycle.
6. The lowest-cost fuel, frequently determined by seasonal demand.
7. Increased security against supply failures and strikes.

Typical performance of gas-burning Diesels shows that less than 5 per cent of the energy is supplied as pilot oil and the rest as natural gas. Three gas-burning Diesels at the Houma, Louisiana, municipal plant generated power during 1950 at a fuel cost of 0.161 cent per kilowatt-hour, when burning natural gas at 12 cents per thousand cubic feet and pilot oil at 9.16 cents per gallon. An average of 10,900 Btu was required per kw-hr, giving a thermal efficiency of 31.2 per cent. This efficiency is based on average load conditions over the year. Higher efficiencies are obtained when the engines are operated at optimum load.

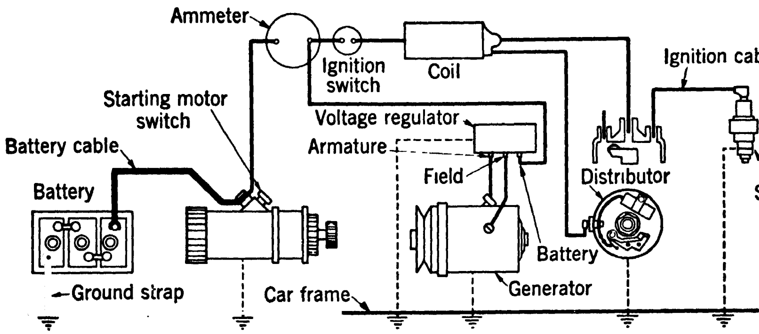
Some advantages of the cylinder arrangement shown in Fig. 263 are (1) few bearings (only two), (2) minimum floor space, (3) easy accessibility for maintenance, (4) light foundation, and (5) a short single-throw crank. The last advantage is most important since crankshafts for large in-line engines may weigh several tons and they are expensive forgings because of multiple cranks, accurate finish, and rifle-drilled oil passage.

282. Spark Control and Ignition Systems. For a variable-speed Otto engine it is necessary to change the time at which the ignition spark occurs if greatest efficiency and power are to be obtained. The fuel should burn as near head-end dead center as possible and at the same time avoid a rate of pressure rise that produces shock or vibration.

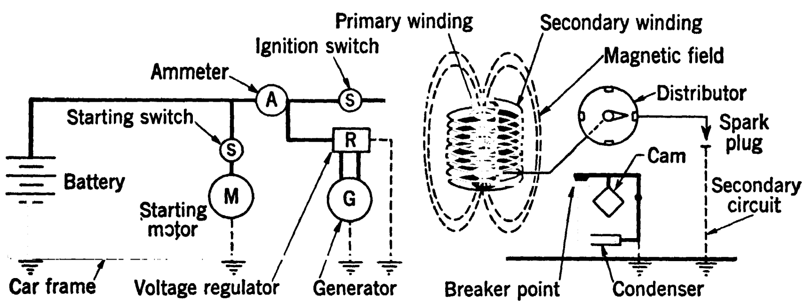
The speed of flame travel in a spark-ignited engine is influenced by many factors, among which are (1) air-fuel ratio, (2) compression pressure, (3) temperature of mixture, (4) turbulence of mixture, (5) kind of fuel, and (6) exhaust gases present. Consequently, each of these factors influences spark timing. Very lean and very rich mixtures burn slowly and require an advanced spark. High turbulence, which occurs with high speed, causes rapid flame travel, and the spark must be retarded for optimum power. Low pressure, low temperature, and dilut-

ing exhaust gases cause a slow flame and necessitate a spark advance.

It has been found practicable to control the spark advance only as a function of speed and intake manifold pressure. Centrifugal weights in the bottom of the distributor head, Fig. 264(*a*), move out as the speed increases, causing the head to rotate with respect to the breaker point, thereby advancing the spark.



(*a*) Component parts



(*b*) Wiring system

FIG. 264. Six-volt battery ignition system.

With partially closed throttle (½ to ¾ setting), the pressure and temperature in the manifold are relatively low; exhaust gas removal is poor; and the carburetor delivers its leanest mixture. All these factors contribute to a slow-burning mixture which requires an advanced spark. A spring-loaded diaphragm connected to the manifold is used to advance the distributor head and to override the centrifugal weights. Figure 265 shows the typical spark-timing characteristics for an automotive engine.

Figure 264 illustrates a modern battery ignition system. The pri-

mary circuit consists of the *battery*, *ammeter*, *ignition switch*, *primary winding*, *breaker point*, and *condenser*. This circuit is completed through the engine block and frame back to the battery. When the breaker point is closed, current flows in the primary circuit, building up from zero to a maximum value, and at the same time a magnetic field is established in the coil. When a spark is desired the breaker point is opened by the distributor cam. This opens the primary circuit but

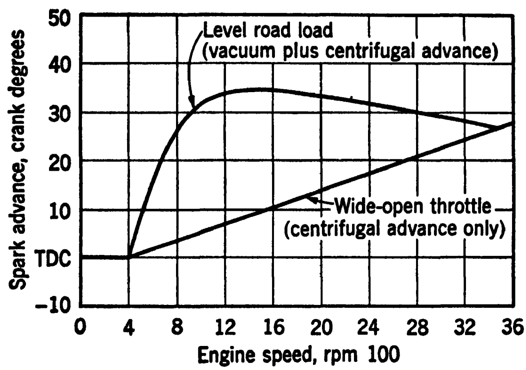


FIG. 265. Spark-advance characteristic for typical automotive engine.

current still tries to flow in the same direction and builds up in the condenser which is in parallel with the breaker point. When the voltage of the condenser becomes greater than that of the battery, the flow is suddenly reversed. This, in turn, reverses the magnetic field in the coil, causing a high voltage to be induced in the secondary winding. Current flows to the distributor, where the arm shorts it to the desired spark plug.

283. Starting Systems. The engine shown in Fig. 254 has a starting system consisting of a two-cylinder auxiliary gasoline engine permanently mounted on the frame; electric and air motor systems are also available for this engine. In operation the starting-engine pinion engages the main-engine flywheel and spins it; thus the lubricant is circulated, and, since the two engines have a common cooling system, the Diesel engine is prewarmed. As a further aid, the exhaust of the starting engine passes through a tube in the inlet manifold to heat the Diesel inlet air.

Many different designs are employed to accomplish the starting operation. Among them are (1) separate engine, as described above; (2) battery and direct-current motor (automotive); (3) compressed-air system; (4) auxiliary combustion chamber; and (5) inertia starter.

284. Engine Cooling. All internal-combustion engines require cooling in certain areas, although ideally they would be more efficient without cooling. Temperatures in the combustion chamber may reach 3500 F, and, although the high temperature exists for only an instant

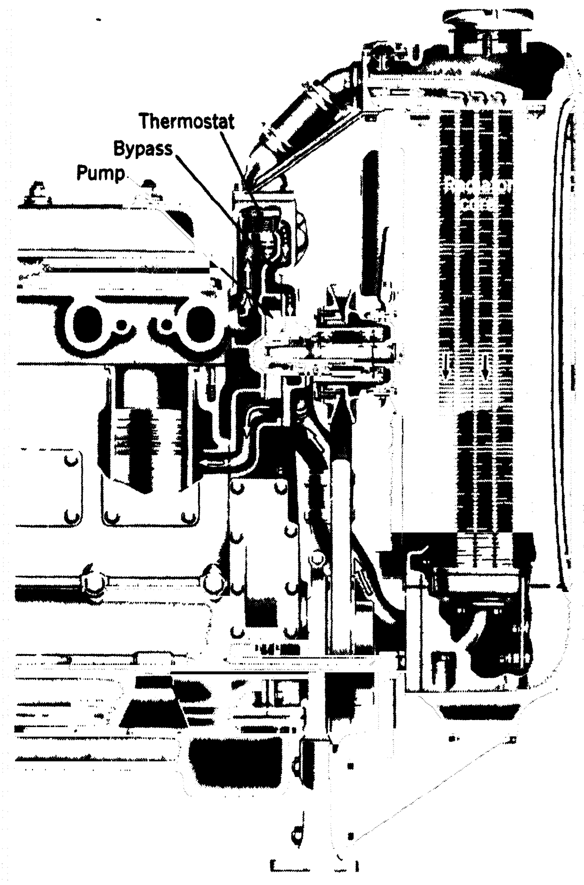


FIG. 266 Liquid cooling system as applied in a Caterpillar engine.

before expansion and exhaust cool the gases, failure would result without some form of cooling—liquid or air. Inadequate cooling may produce: (1) burning of pistons or exhaust valves, (2) excessive detonation in gasoline engines, (3) seizure of piston due to expansion, (4) lubricant burning, and (5) melting of a bearing material.

Figure 266 shows a liquid cooling system as applied to a Diesel engine. The thermostat maintains the desired temperature around the

engine block before the coolant is allowed to pass through the radiator —a circulation by-pass permitting flow through the engine when the thermostat is closed. Not only is a minimum temperature gradient across the block desirable but the flow of water must be so controlled that temperatures will be high enough throughout the engine system to prevent cylinder wear from low-temperature corrosion. The system is designed to prevent entrapping of air and to direct the coolant flow against critical parts, such as the exhaust valve seats.

The energy derived from combustion leaves the engine as (1) shaft work, (2) exhaust losses, or (3) cooling-system losses. All friction losses within the engine eventually appear as heat and are removed largely through the cooling system.

285. Lubrication and Oil Purification. A lubrication system is expected to (1) minimize friction forces, (2) prevent metal-to-metal contact of moving parts, (3) minimize wear, (4) prevent oxidation of finished parts, (5) remove carbon, dirt, or metal particles, (6) provide internal cooling for the engine, and (7) provide seal for gases at piston rings. If an engine is to be lubricated by a single system, the oil selected must be a compromise to cover a wide range of speeds, loads, and temperatures. Some desirable properties of a lubricant are (1) low viscosity, (2) no change in viscosity with temperature, (3) chemical stability, (4) detergent action, (5) absence of volatility, and (6) non-inflammability.

Low-viscosity oil reduces engine friction and improves performance; however, the viscosity must not be so low that bearing failure results or excessive oil is pumped past the piston rings. Some oil must be expected to pass the upper piston ring and be burned if the upper ring is to be lubricated.

The rate of oil consumption is a function of (1) engine speed, (2) oil temperature, (3) oil viscosity, (4) oil volatility, (5) piston and ring design, and (6) condition or state of wear of cylinder and rings. The three ways by which oil may escape from an engine are burning on the cylinder walls, leakage past gaskets and shaft seals, and vaporization through the crankcase breather. High-speed operation accelerates all three. The standard viscosity classification of motor oils is shown in Fig. 267 in which a given number indicates a certain range of viscosity as measured on a Saybolt Universal viscosimeter, Art. 86. Thus SAE[1] 20 oil has a viscosity of 103 sec at 150 F and a viscosity of 320 sec at 100 F. Some synthetic oils such as Silicones have a lesser tendency to change viscosity with temperature.

Most oils are now treated with additives and are given a trade name

[1] Society of Automotive Engineers.

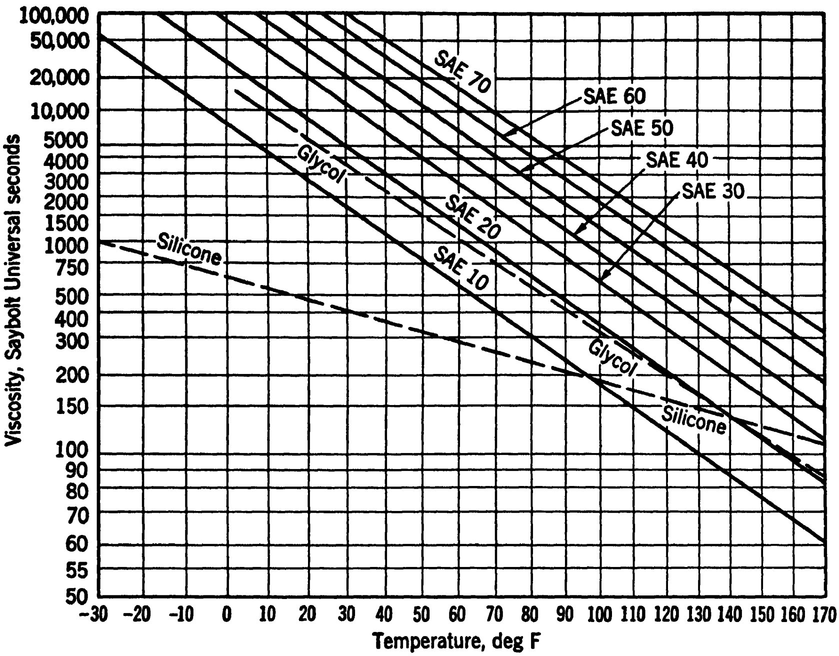


FIG. 267. ASTM viscosity chart.

or sold as "compound" or "heavy duty"oils. Such oils provide some or all of the following characteristics: (1) detergent action, (2) anti-corrosion, (3) anti-oxidation, (4) viscosity improvement, (5) foam prevention, (6) pour-point depressant, (7) film-strength improvement, and (8) extreme pressure (EP) property, as in hypoid gear lubricants.

Lubricating oil does not "wear out" but becomes contaminated from such sources as road dust, fuel blown past the rings, combustion products, metal particles, and scale. The various methods of cleaning used oil are (1) gravity settling, (2) centrifuge, Fig. 268, and (3) filtration; filtration may be (*a*) mechanical (wire-mesh strainer, edge-type copper ribbons), (*b*) adsorbent (charcoal, bone black, activated clay), or absorbent (cloth felt, flannel, paper, cotton).

286. Supercharging. The output of a naturally aspirated cylinder is limited by the oxygen it can obtain. By means of a supercharger more air can be forced into the cylinder, more fuel can be burned, and a higher mean effective pressure produced. Centrifugal and rotary positive displacement superchargers are air compressors and were discussed in Chapter 14. The rotary positive displacement type, Fig. 269, is driven from the engine shaft, whereas the centrifugal is frequently

driven by an exhaust turbine and is called a turbocharger. Figure 270 shows a turbocharger as used on typical stationary engines.

The effects of supercharging on the ideal four-stroke-cycle Diesel may be seen in Fig. 271, in which air at point 11 is isentropically compressed to point 1, where it enters the engine intake manifold. From point 1

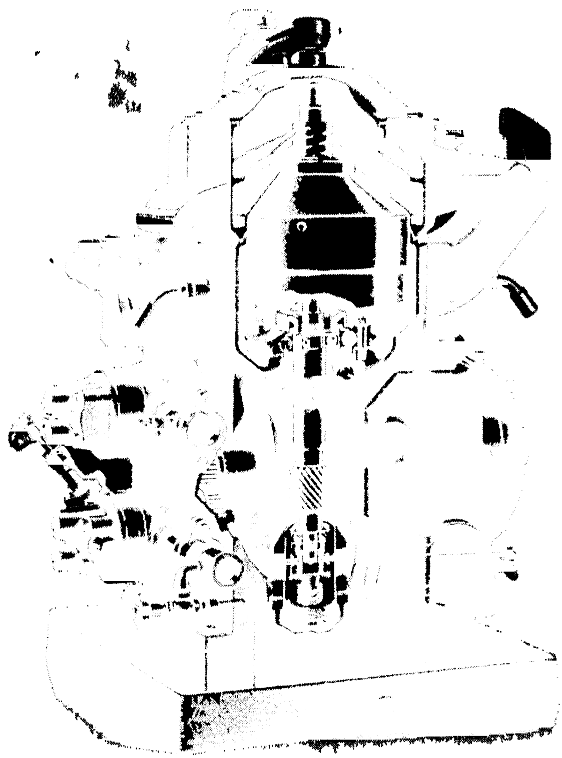


FIG. 268 De Laval "Uni-Matic" oil purifier.

the air goes through the usual Diesel cycle 1–2–3–4–1. At point 4 it leaves the cylinder by way of the exhaust valves which restrict the flow and cause a large pressure drop. If the intake manifold pressure and the exhaust manifold pressure are equal, the exhaust gases arrive at point 8 after irreversible expansion from point 4; thus the gases would do work on the turbine as they expand to atmospheric pressure at point 9. With these assumptions of equal pressures in intake and exhaust manifolds, the work of the exhaust turbine would be area 6–7–8–9–6. The compressor work done on the air in supercharging

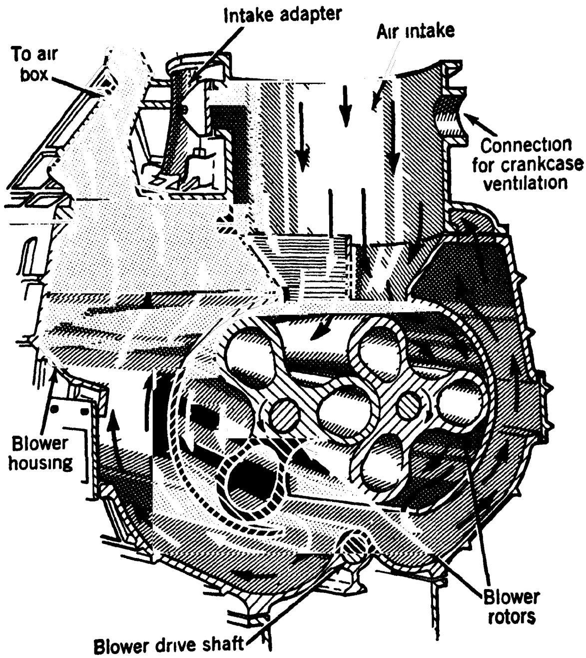


FIG 269 General Motors blower

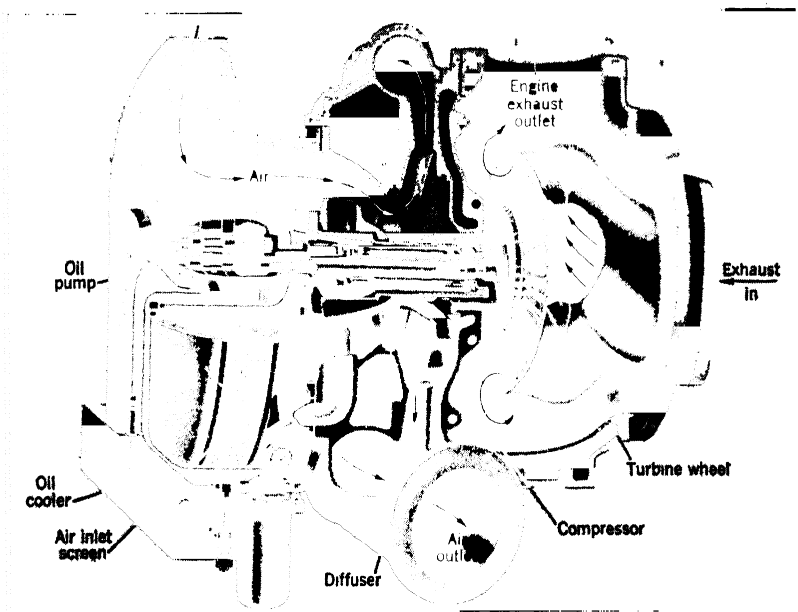


FIG 270 Cutaway view of Elliott-Buchi turbocharger

would be the area 6–7–1–11–6. The indicated work from the engine would be area 1–2–3–4–1. The difference between turbine- and compressor-work areas would theoretically be available for useful shaft

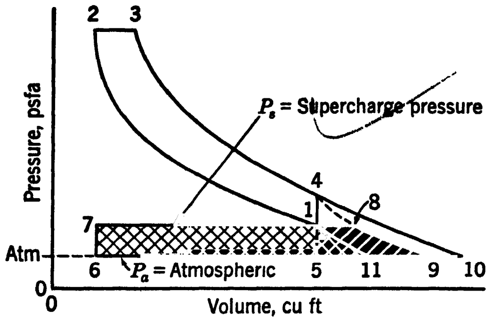


FIG. 271. Supercharged Diesel cycle.

output; however, inefficiencies of the compressor and the turbine more than consume this difference, and both the intake and exhaust manifold pressures stabilize at some pressure level dependent upon the engine load and the efficiencies of the compressor and the turbine.

By use of supercharging, power increases as high as 50 per cent of the unsupercharged power are possible with no change in thermal efficiency. The cylinders are relieved of the exhaust and suction work which would appear as part of the frictional losses in naturally aspirated engines. Pressures can be held constant, and the engine can continue to deliver sea-level power at high altitudes. Four-stroke-cycle engines are more adaptable to supercharging than two-stroke-cycle engines.

287. Scavenging. Two-stroke-cycle engines require that the products of combustion be forced from the cylinder by some type of blower or compressor. The two-stroke-cycle Diesel engine is adaptable to this scavenging process since the air does not carry fuel with it as in a gasoline engine. The pressure used for scavenging and supercharging two-stroke-cycle Diesel engines seldom exceeds 4 psig since at approximately that value the power required by the blower exceeds the gains derived.

Figure 272 shows a Diesel engine designed for *cross-scavenging* and equipped with automatic nonreturn valves. *Through-scavenging*, illustrated in Fig. 273, is obtained by venting the exhaust at one end of the cylinder and admitting air at the other. The valves, or ports, may be unsymmetrically timed so that the exhaust port leads the inlet port, or valve, in both opening and closing, thus permitting any degree of supercharge. The opposed-piston engine of Fig. 273 is adaptable for use as a Diesel, a dual-fuel, or as a spark-ignited gas engine. Such a

two-stroke-cycle engine does not need valves, push rods, rocker arms, or cylinder head; however, two crankshafts, two sets of pistons, and a vertical interconnecting shaft are required. An 8⅛-in. bore by 10-in. stroke engine of this type, when operating as a Diesel, delivers 160 bhp per cylinder at 720 rpm.

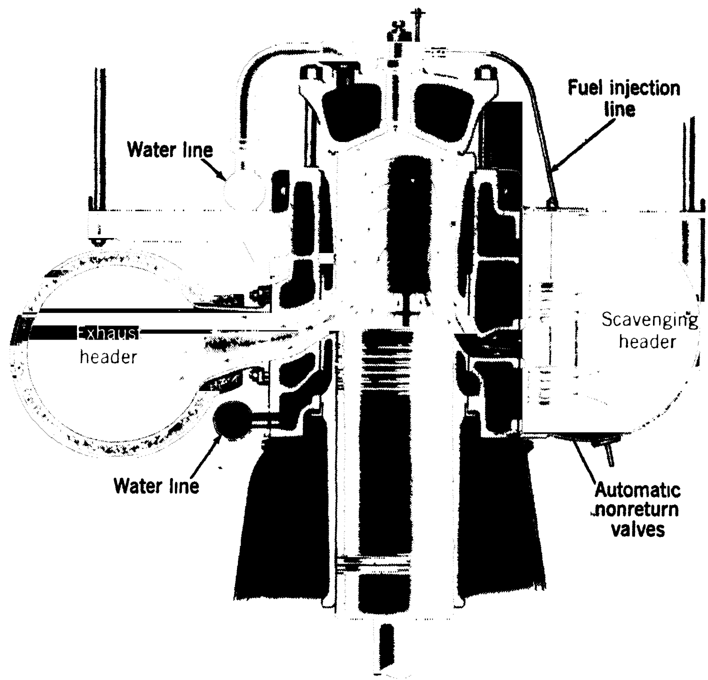


FIG. 272. Nordberg two-stroke-cycle Diesel engine.

288. Carburetion. The functions of a carburetor are (1) to control the amount of oxygen and the amount of fuel entering the manifold, (2) to provide the desirable ratio of fuel to air through various ranges of speed and load, and (3) to aid in mixing the fuel and air. A Venturi tube is used in which increased air flow causes a depression at the throat for aspirating a flow of gasoline. Throttle valve, choke, acceleration pump, and float chamber with metering orifice are auxiliary components. The throttle controls the amount of mixture (air and fuel) entering the cylinder; and the choke is used to restrict the air flow, thereby controlling mixture richness. For Otto-cycle engines the entire breathing system—air filter, carburetor, intake manifold, and

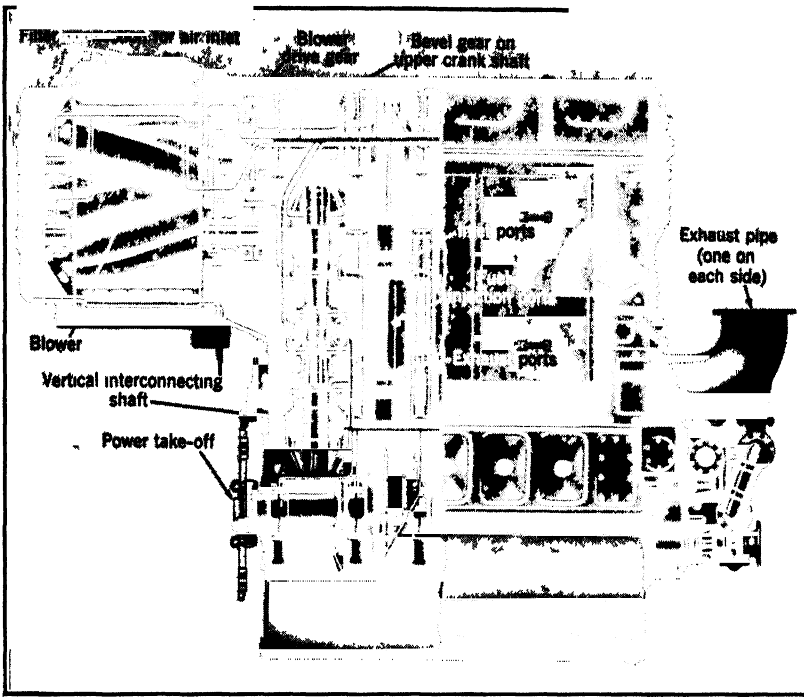


FIG 273 Fairbanks-Morse opposed-piston engine

valve passages—should offer the minimum flow resistance if good volumetric efficiency is to be obtained. Figure 274 shows schematically a breathing system for a carburetted engine.

Figure 275 shows an ideal performance curve of a carburetor for a variable-speed engine As the throttle is changed from completely closed to completely open, the air-fuel ratio should vary approximately as indicated During starting or idling of a cool engine it is necessary to have a rich mixture because of (1) poor vaporization of fuel, (2) poor distribution of fuel, (3) high dilution from previous cycle, (4) poor mixing, (5) low temperature, and (6) low compression.

Figure 276[2] illustrates the *idle jet* which by-passes the throttle valve and delivers a rich mixture only when the throttle is nearly closed As the throttle opening approximates 25 per cent, the flow through the idle jet is reduced to zero. This gives a lean mixture throughout the usual range of operation (25 to 80 per cent of capacity), as shown in Fig. 277.

[2] Figures 276, 277, 278, and 279 are included through the courtesy of Carter Carburetor Co , St Louis, Missouri.

Enriching the mixture near wide-open throttle is usually done by withdrawing a tapered needle called an *economizer* from the main fuel orifice. This enlarges the passage and allows more fuel to flow, Fig. 278.

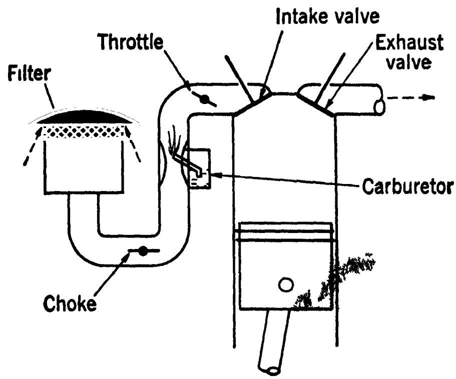


FIG. 274. Breathing system for carburetted engine.

To provide a quick response on opening of the throttle, an acceleration pump in automotive carburetors, Fig. 279, delivers a jet of gasoline as the

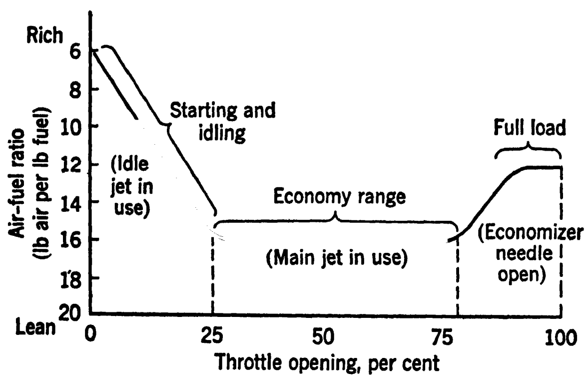


FIG. 275. Idealized carburetor characteristic.

throttle is opened. Without it, air flow would exceed gasoline flow and the engine would starve for a brief interval. The carburetor Venturi is the basic device for proportioning the flow of gasoline to that of air.

In the economy range, Fig. 275, the efficiency is greatest with a lean mixture for the following reasons:

1. All fuel is burned since extra oxygen is present.
2. Specific heat of the mixture is less.

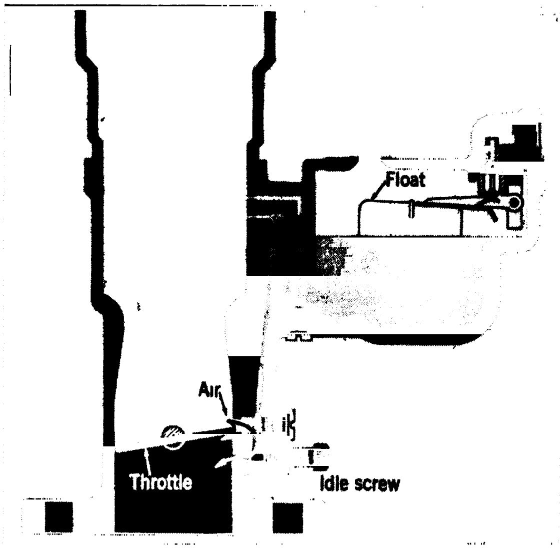


FIG 276 Carburetor idle circuit, low speed

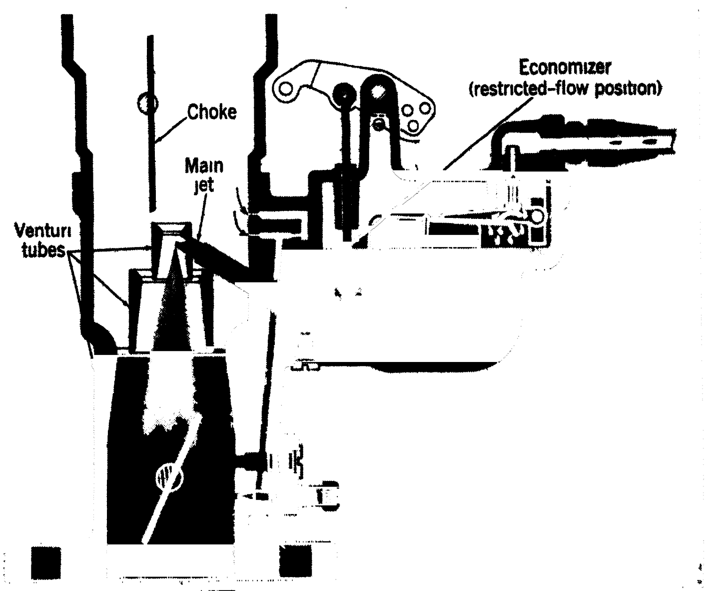


FIG. 277. Carburetor economy circuit, normal operation.

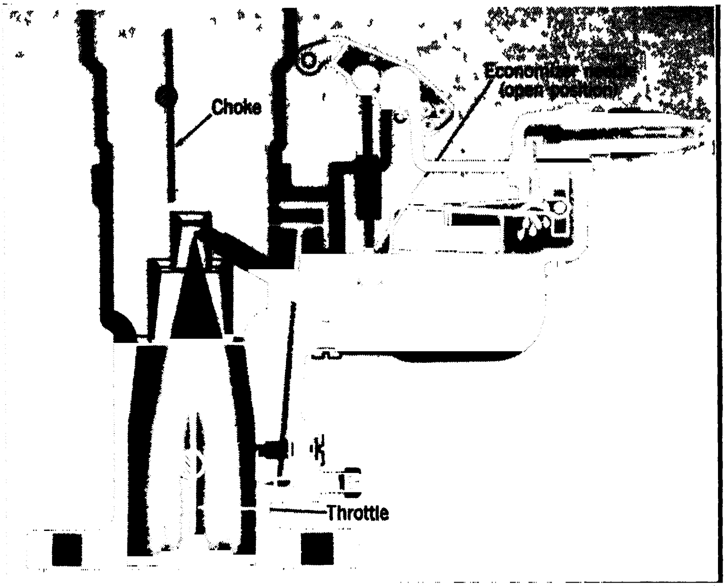


FIG 278 Carburetor power circuit, wide-open capacity.

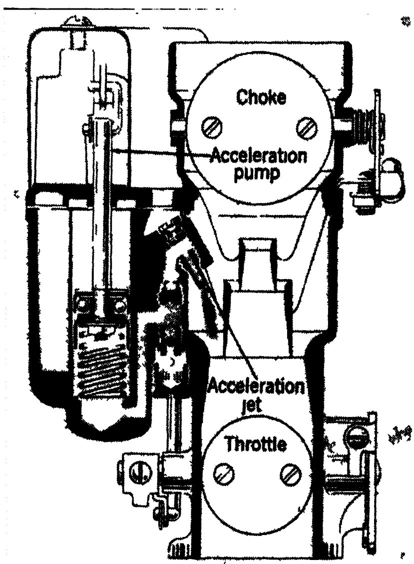


FIG. 279 Carburetor pump circuit, acceleration,

3. Less heat transfer occurs since temperatures are lower.
4. Dissociation is less for the same reason.
5. Distribution is better since evaporation is accelerated.
6. Mixing is better since gas and air velocities are higher.

When maximum power is needed it has been found desirable to enrich the mixture for the following reasons:

1. All oxygen is used.
2. Flame speed is increased with more burning at head-end dead center.
3. Excess fuel provides internal cooling.
4. Valve burning is prevented by using all the oxygen.

289. Fuel Injection. Air injection was one of the first methods for delivering the small amount of fuel required per cycle against the high compression pressure of the Diesel engines (400 to 600 psig). A three-stage air compressor was used to deliver air at approximately 1000 psig, and a metering pump placed the oil just ahead of the air inlet valve. Good combustion was obtained but compressor maintenance proved to be a great disadvantage.

Liquid injection uses some form of small high-pressure pump whose capacity can be varied by rotating the plunger in the barrel. This action varies the opening and closing of the ports and consequently the amount of fuel delivered. Figure 280 shows a cross section of a pump of this type. Fuel from the supply pump is forced through a filter made of cotton string tightly wound on a screen core on which is a layer of fine filter paper. The clean oil fills the manifold at (1) and the passages leading to the ports at (2). As the injection pump plunger is lifted by the cam, Fig. 281, fuel delivery begins when the plunger covers the port and continues until the port is uncovered by the spiral scroll. The plunger is rotated by the pinion (3) and rack (4), which is under governor control, Fig. 280. The oil under high pressure is then delivered through a thick-walled tube to the injection valve in the cylinder head; this valve, which contains a spring-loaded check valve, designed to open at a predetermined value, is made of precision parts to control accurately the fuel flow and oil-spray pattern into the precombustion chamber.

290. Combustion Chambers. In spark-ignited engines the desire to obtain high efficiency has led gradually to a combustion chamber that is hemispherical in shape with the valves mounted overhead as shown in Fig. 282(*a*). Other typical combustion-chamber designs are illustrated in Fig. 282, parts *b* and *c*. Efficiency increases when the heat-transfer surface is a minimum compared with the contained volume. To prevent detonation, the shortest possible flame travel is necessary, and the spark plug is placed at the apex of the hemisphere. If the power-weight ratio, efficiency, and reliability are sufficiently important,

two ignition systems may be used, and the two spark plugs per cylinder cut in half the length of flame travel.

Diesel combustion chambers may be broadly classified as turbulent or restricted, Fig. 280, and nonturbulent or open, Fig. 272. The various

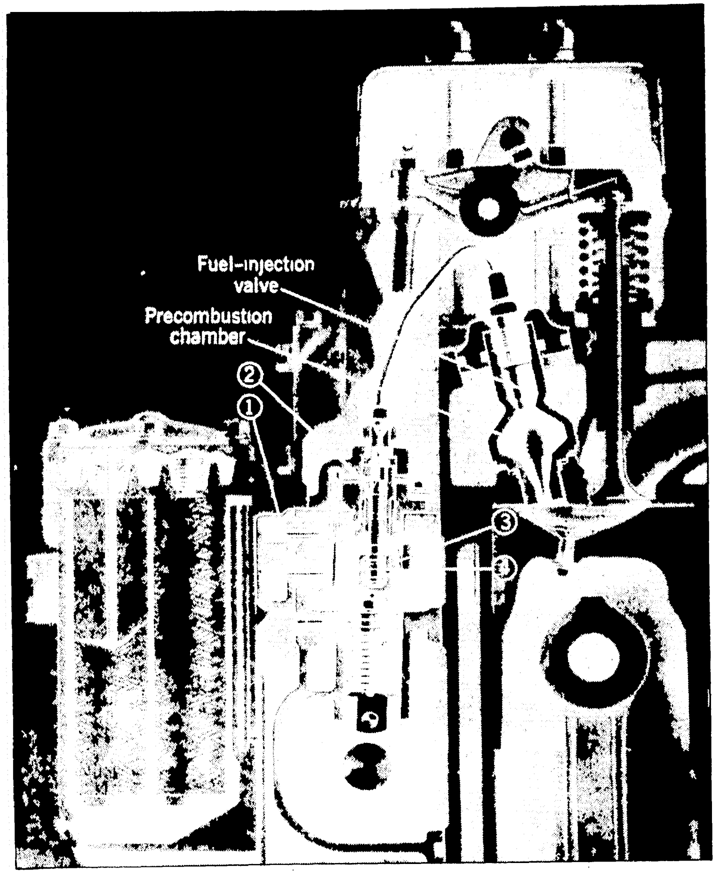


FIG. 280. Caterpillar fuel injection system.

designs for producing turbulence in high-speed Diesel engines spring from the need for thorough mixing of the fuel and air in order to complete combustion in the short time available The turbulent or precombustion chamber, Fig. 280, greatly assists in the mixing operation, provides cleaner burning and better atomization of a variety of fuels, and permits operation at a wide range of speeds and loads without fouling The operation is based on the fact that all fuel oil contains some light hydrocarbons that will vaporize at a fairly low temperature. The burning of

these lighter parts in the precombustion chamber provides the energy whereby the rest of the charge is forced out violently into the cylinder with high turbulence and good mixing. The precombustion chamber

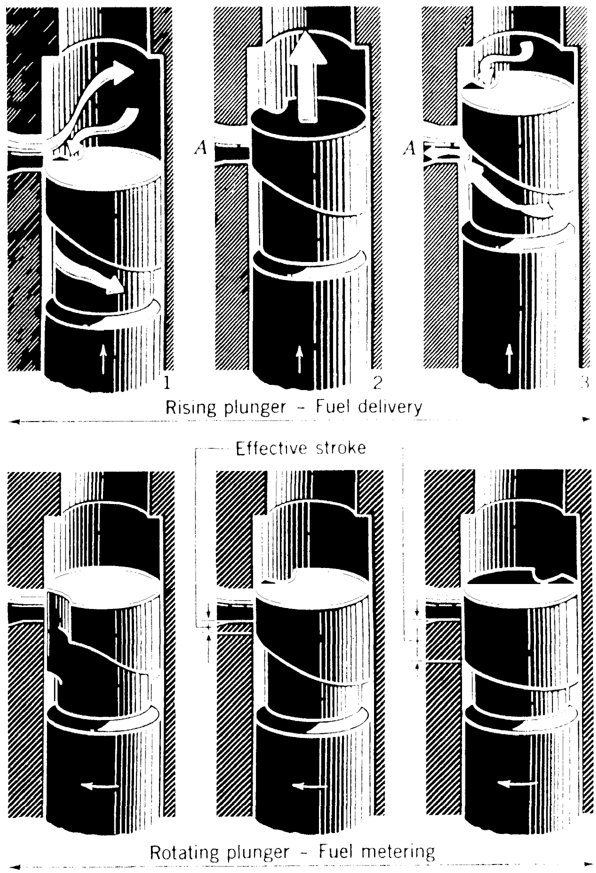


FIG. 281. Operation of Caterpillar fuel injection pump.

design permits idling for prolonged periods of time without clogging of the injection nozzle, and it also permits the use of a simpler one-orifice injection nozzle.

The open combustion chamber which has less turbulence must rely more on the injection nozzle to direct the fuel to the air. Consequently, it is generally a multiple-orifice type. Scavenging efficiency may be

better and heat transfer less, owing to a simpler shape of combustion chamber, as illustrated in Fig. 272.

Adequate mixing is no problem in the Otto engine since, beginning at the carburetor, most of the gasoline evaporates in the manifold; passing through the intake valves the gasoline is further heated and

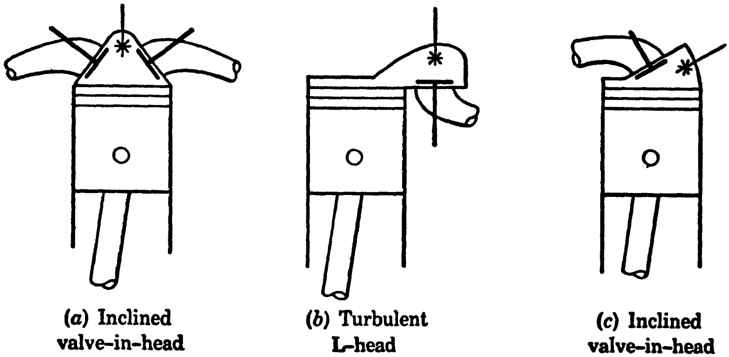


FIG. 282. Typical spark-ignited engine combustion chambers.

mixed as the piston moves through the suction and compression strokes. The resulting homogeneous mixture of fuel and air is ready to be ignited by the spark; therefore the Otto engine is capable of operating at high speeds—4000 rpm and more.

291. Governing. In spark-ignited engines control is accomplished by means of a throttle which consists usually of a thin plate or butterfly valve in the intake manifold immediately following the carburetor, Fig. 278. The amount of opening of this valve (by foot control or accelerator pedal in automobiles) determines how much mixture may enter the cylinder. This, in turn, determines the mean effective pressure and consequently the power and speed developed within the engine. The positioning of the throttle may be controlled by rotating centrifugal weights as a function of speed; such an arrangement is called a *speed governor*. For certain applications the throttle may be positioned as a function of the load while the speed is held constant.

In Diesel engines only the fuel flow is regulated, the air flow being unrestricted. Since the Diesel engine does not rely on fixed proportions of air and fuel for combustion, any fuel-air ratio may be used up to a point where the oxygen is all consumed.

292. Engine Testing. The two usual conditions under which engines are operated are: (1) variable speed with variable load and (2) constant speed with variable load. The first is found in such

applications as automobiles, railway engines, and tractors. The second is found in a-c generator drives.

In testing, the information desired involves power and efficiency over the allowable speed and load range. Determinations of ***indicated, brake,*** and *friction horsepower* together with *mechanical efficiency*, as explained in Arts. 11 to 14, are usually made.

In an engine performance test the following data are usually required:

1. *Fuel consumption.* The weight or volume of fuel used is determined for a given time while the engine is operating at the desired load and speed conditions. For gas engines an accurate gas meter is employed. Since the heating value of the fuel must be determined, the temperature and pressure of the gas in the meter must be taken. Usually "wet" meters are employed so that the gas is known to be saturated.
2. *Density of fuel.* The API gravity of liquid petroleum is taken as a measure of the density and heating value of the fuel, Art. 85.
3. *Engine speed.*
4. *Net load or torque developed.*
5. *Coolant temperature in and out of the engine.*
6. *Lubricating oil temperature.*
7. *Exhaust back pressure.*
8. *Density of air.* The barometric pressure and wet- and dry-bulb temperatures of the air are necessary for correcting the results to standard conditions. The indicated power and indicated torque vary directly as the barometric pressure and inversely as the square root of the absolute temperature. Thus an engine at sea-level conditions would have an advantage over one operating above sea level, because of the greater density of the oxygen. Likewise an engine breathing cool air would have greater capacity to burn fuel and produce high mean effective pressures and consequently higher power. The correction factor for nonstandard conditions is

$$f = \frac{B_{std}}{B_{dry}}\sqrt{\frac{T_{obs}}{T_{std}}} \tag{314}$$

where T_{obs} = observed carburetor air temperature, deg F abs.

B_{std} = 29.92 in. mercury.

$B_{dry} = B_{obs} - p_v$, in. mercury.

B_{obs} = barometric pressure observed, in. mercury

p_v = vapor pressure $= p_w - \dfrac{B_{obs}(t_d - t_w)}{2700}$, in. mercury

p_w = vapor pressure at t_w from steam tables, in. mercury

t_d = dry-bulb temperature, deg F.

t_w = wet-bulb temperature, deg F.

The correction is applied to the *indicated* values of horsepower, torque, and mean effective pressure of Otto engines when operated at wide-open throttle. When the effect of vapor pressure is neglected $B_{dry} = B_{obs}$.

Other terms usually calculated in engine performance are (1) brake mean effective pressure, (2) friction horsepower, Art. 12, (3) mechanical efficiency, Art. 13, (4) indicated thermal efficiency, (5) brake thermal efficiency, and (6) volumetric efficiency.

The *brake mean effective* pressure is the mean effective pressure (*mep*) which, if acting on the piston, would develop power equivalent to the brake horsepower; it is calculated from the equation

$$bmep = \frac{bhp \times 33{,}000}{lAN'} \text{ psi} \tag{315}$$

where l = length of stroke, ft.
A = cross sectional area of cylinder, sq in.
N' = number of power impulses per minute.

The brake mean effective pressure affords an accurate means for comparison of engine performance and is used as such in preference to the indicated mean effective pressure. Because of the variable factors that influence combustion, such as the amount of burned gases left in the cylinder and the air-fuel ratio, the successive cycles of an internal-combustion engine may vary considerably. This variation is eliminated, however, in the mean effective pressure based on the brake horsepower.

The *indicated thermal efficiency* shows what fraction, or percentage, of the heat supplied was transformed into indicated (developed) horsepower in the engine cylinders.

$$e_i = \frac{2545 \times ihp}{m_f \times H_f} \tag{316}$$

where ihp = indicated horsepower of engine.
m_f = fuel supplied per hour, in cu ft for gaseous fuels, in lb for liquid fuels.
H_f = higher heating value of fuel supplied, Btu per cu ft or per lb, as required.
2545 = 1 horsepower-hour equivalent, Btu.

The *brake thermal efficiency* shows what fraction, or percentage, of the heat supplied was actually delivered by the engine shaft.

$$e_b = \frac{2545 \times bhp}{m_f \times H_f} \tag{317}$$

where bhp = brake horsepower of engine. Other symbols are the same as for equation 316.

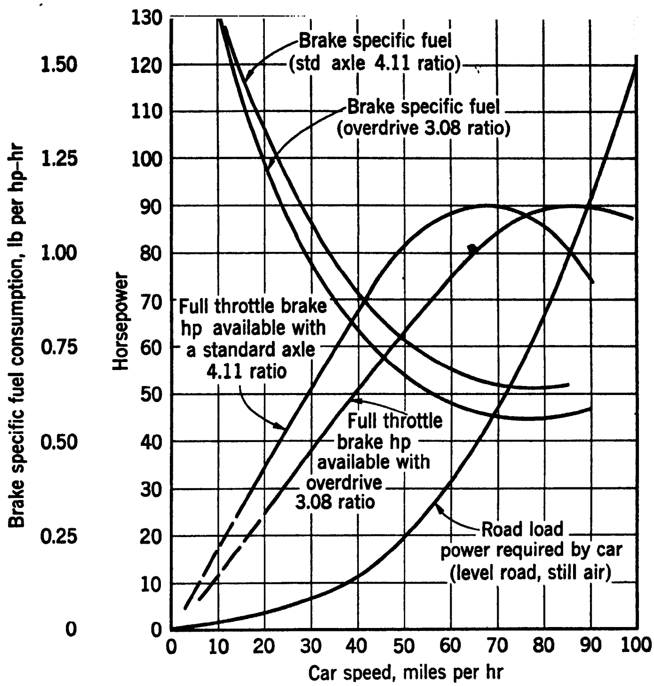


FIG. 283. Performance curves for modern six-cylinder automotive engine.

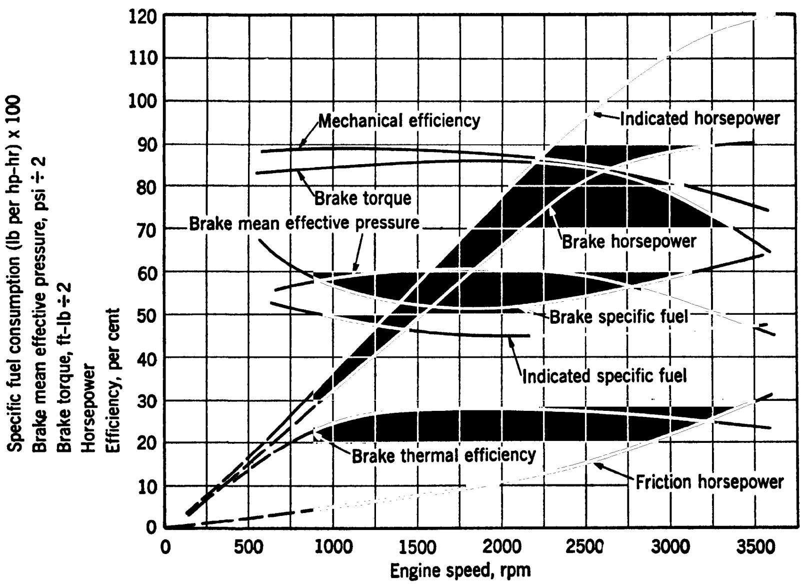


FIG. 284. Performance curves for modern six-cylinder automotive engine.

The *volumetric efficiency* is the ratio of the weight of air which enters or is forced into the cylinder during the suction stroke to the weight of free air equivalent to the piston displacement.

$$e_v = \frac{m_a}{m_t} \tag{318}$$

where m_a = actual weight of air inducted, lb.
m_t = weight of free air equivalent to the piston displacement, lb.

With this information the performance characteristic of the engine may be calculated. Figures 283 and 284[3] show some performance curves for a typical automotive engine when operated at wide-open throttle and varying speed. The curves have been corrected to standard conditions. This four-stroke-cycle engine was the overhead-valve type with 3½-inch bore, 3¾-inch stroke, and a rating of 90 bhp at 3300 rpm. The compression ratio was 6.6 : 1.

PROBLEMS

1. Determine the air-standard Otto-cycle efficiency for the following compression ratios: 8, 10, and 12.

2. Determine the air-standard Diesel-cycle efficiency for the following compression ratios: 8, 10, and 12. Fuel is injected for 10 per cent of the stroke in each case.

3. An engine operates on an air-standard cycle in which there is no compression. Combustion is at constant volume and expansion is isentropic to the atmospheric pressure line. The clearance volume is 0.25 cu ft. If the pressure is 14.7 psia and the temperature 520 F abs at the beginning of combustion and the maximum pressure in the cycle is 100 psia, find (*a*) the ideal cycle efficiency, (*b*) power developed for 200 cycles per min, and (*c*) the mean effective pressure.

4. Calculate the clearance volume in per cent of the displacement for an air-standard Otto cycle having an efficiency of 50 per cent. 21.4 per cent, *Ans.*

5. In an ideal air-standard Otto cycle the pressure, volume, and temperature at the beginning of compression are 14.0 psia, 0.5 cu ft, and 100 F, respectively. At the end of compression the pressure is 275 psia. Find the theoretical horsepower developed for 500 cycles per min and 10 Btu supplied per cycle.

6. An engine operates on the air-standard Otto cycle with a compression ratio of 8:1. Pressure and temperature at the inlet are 14.7 psia and 90 F, respectively. $k = 1.4$. Find (*a*) the temperature at the end of compression, (*b*) the air-standard efficiency of the cycle, (*c*) maximum pressure if 70 Btu are added per cu ft of air displaced (constant specific heat $c_v = 0.171$), (*d*) energy rejected per cycle, and (*e*) net work done per cycle in Btu.

7. A single-cylinder single-acting four-stroke Diesel-cycle engine having a 12-in. bore and 18-in. stroke operates at 200 rpm. The mean effective pressure as obtained from the indicator card is 96 psi. Other test data are as follows: length of brake arm, 60 in.; fuel used, 20 lb per hr; heating value of fuel oil, 19,500 Btu per lb; net

[3] Figures 283 and 284 are included through the courtesy of I. D. Rea, Urbana, Illinois.

weight on brake, 220 lb. Find (*a*) indicated horsepower, (*b*) brake horsepower, (*c*) friction horsepower, (*d*) mechanical efficiency, and (*e*) brake thermal efficiency.

8. The average indicator card from a nine-cylinder four-stroke cycle 5.5-in. bore by 6-in. stroke engine is 4 in. long and has an area of 6 sq in. The maximum card height is 3.5 in. and represents a maximum pressure rise of 666 psi. Find (*a*) indicated mean effective pressure, (*b*) indicated horsepower of engine at 1800 rpm, (*c*) indicated torque, (*d*) brake mean effective pressure if mechanical efficiency is 87 per cent, and (*e*) brake torque.

(*a*) 285.8 psi, (*b*) 833 ihp, (*c*) 2430 ft-lb, (*d*) 248.5 psi, (*e*) 2120 ft-lb, *Ans.*

9. The following data were obtained during a test of a six-cylinder four-stroke Otto-cycle engine: speed, 2000 rpm; bore, 3.5 in.; stroke, 3.75 in.; specific fuel consumption based on the brake horsepower, 0.66 lb per bhp-hr; brake arm, 1.05 ft; heating value of fuel, 20,000 Btu per lb; brake torque, 133 ft-lb; friction torque, 40 ft-lb; room temperature, 90 F; barometer, 27.50 in. mercury. Find (*a*) brake horsepower, (*b*) brake mean effective pressure, (*c*) indicated horsepower, (*d*) indicated mean effective pressure, (*e*) brake thermal efficiency, and (*f*) indicated thermal efficiency.

10. The following data were obtained during a full-load test on a two-cylinder single-acting four-stroke Diesel-cycle engine: 11-in. bore by 15-in. stroke; average rpm, 385; length of Prony brake arm, 36 in.; net weight on scales (due to torque), 500 lb; total fuel used in 15-min run, 11.5 lb; heating value of fuel oil, 19,500 Btu per lb; mean effective pressure from indicator cards (average of each cylinder), 90 psi. Find (*a*) brake horsepower, (*b*) brake specific fuel consumption, (*c*) thermal efficiency, (*d*) indicated horsepower, (*e*) friction horsepower, and (*f*) mechanical efficiency.

11. A six-cylinder four-stroke Otto-cycle engine of 3-in. bore and 3¼-in. stroke has a brake thermal efficiency of 26 per cent. Determine the shaft horsepower that the engine can develop when operating at 3600 rpm, it if is supplied with an air-gasoline mixture having a heating value of 84 Btu per cu ft and if volumetric efficiency is 90 per cent.

12. Find the bore and stroke in inches of a six-cylinder four-stroke-cycle gasoline engine which delivers 120 bhp at an average piston speed of 2500 fpm. Assume: volumetric efficiency, 85 per cent; mechanical efficiency, 80 per cent; gasoline, 20,000 Btu per lb; theoretical air-fuel ratio, 15.0 lb per lb; excess air, 20 per cent; brake thermal efficiency, 23 per cent; stroke-bore ratio, 1:1; gasoline vapor twice as heavy as air; air and gasoline vapor at 14 psia and 180 F at end of suction stroke.

4.47 in., *Ans.*

13. If N_2 in the air (79 per cent by volume) is replaced by helium ($k = 1.67$), what would be the efficiency of the ideal Otto cycle with a compression ratio of 8:1? $c_p = 1.25$, $c_v = 0.75$ for helium.

14. Determine the quantity of fuel oil in pounds and in cubic inches to be injected per cylinder per cycle for a 65-bhp four-stroke Diesel-cycle six-cylinder engine running at 2500 rpm. The brake specific fuel consumption is 0.4 lb per bhp-hr, and the specific gravity of the oil is 0.76 referred to water at 60 F.

15. An automobile engine at wide-open throttle developed 90 bhp under test. The barometric pressure was 28.6 in. mercury. The room temperature was 89 F; the wet-bulb temperature, 72 F; and mechanical efficiency of this engine, 86 per cent. Determine the brake horsepower that would be developed at standard conditions, 29.92 in. mercury and 60 F.

16. Tests on a six-cylinder four-stroke-cycle Diesel engine with 4¾-in. bore and

6-in. stroke gave the following results: speed, 1190 rpm; net force on dynamometer, 180 lb; torque arm, 24 in.; fuel used per hour, 31 lb; net force on dynamometer due to motoring, 22 lb at 1190 rpm; heating value of fuel, 19,800 Btu per lb; air-fuel ratio, 20:1; barometric pressure, 14.3 psia; ambient temperature, 95 F; wet-bulb temperature, 80 F. Find (*a*) actual brake horsepower, (*b*) friction horsepower, (*c*) indicated horsepower, (*d*) brake specific fuel (pounds per brake horsepower-hour), (*e*) indicated specific fuel consumption, (*f*) brake mean effective pressure, (*g*) indicated mean effective pressure; (*h*) brake torque, and (*i*) indicated torque.

17. For the engine in problem 16 determine the following efficiencies: brake thermal, indicated thermal, mechanical, and volumetric. Determine the Btu supplied per cubic foot of suction displacement.

18. A four-cylinder four-stroke-cycle Diesel engine, 16-in. bore by 20-in. stroke, is assumed to operate on the air-standard cycle. Compression ratio, 14:1; volumetric efficiency, 100 per cent; inlet conditions, 14.696 psia and 80 F. If 90 per cent of the oxygen is consumed by burning $C_{12}H_{26}$ (heating value, 20,550 Btu per lb) at constant pressure, find the theoretical horsepower when the engine is operating at 200 rpm.

19. If an exhaust-driven supercharger and an inlet air cooler are installed on the engine of problem 18, the inlet air conditions are changed to 20.7 psia and 80 F. Determine the theoretical indicated power that the engine would develop.

CHAPTER 17

Mechanical Refrigeration

293. Applications. The fields of mechanical refrigeration or cooling include household refrigerators; the quick freezing of meats, fruits, and vegetables; cooled spaces for the storage of perishables under many different conditions; ice making; industrial processes which require refrigeration; and the cooling and dehumidification of air in special phases of air conditioning.

294. General Process. The assemblage of equipment necessary for mechanical refrigeration is often classed as a heat pump. The process involves the absorption of heat from a material to be cooled by a working medium at a low-temperature level and the discharge of this heat, from the system, at a higher potential of temperature.

295. Reversed Carnot Cycle. The functioning of a heat engine under the different phases of an ideal Carnot cycle was explained by the Ts diagram of Fig. 34 in Art. 48. The temperature-entropy diagram of Fig. 285 shows the processes of a working medium (refrigerant) being completed in the reverse order of those of the power cycle of Fig. 34. In the reversed cycle, Fig. 285, work is done upon the refrigerant to elevate its absolute temperature from T_2 to T_1 deg F.

The following sequence of events occurs when work is done upon a refrigerant under the conditions of a reversed Carnot cycle. (1) Isentropic expansion of the refrigerant occurs between a and b with a drop of temperature from T_1 to T_2 as the pressure of the working medium falls. (2) Heat Q_2, represented by area b–c–f–e–b, is absorbed by the working medium, at a temperature T_2, when by use of a piece of heat-transfer apparatus some fluid such as either air or a liquid is cooled. (3) The pressure of the refrigerant is increased and its temperature elevated to T_1 by isentropic compression between state points c and d. The work of compression, $-W/J$ in Btu per lb, is proportional to the area c–d–a–b–c. The minus sign of $-W$ merely indicates that work is done upon the refrigerant and has no other significance here. W represents foot-pounds of work per pound of refrigerant, and J is the mechanical equivalent of heat 778 ft-lb per Btu. (4) From d to a at an absolute

temperature T_1 and constant pressure heat flows to a medium (either air or a liquid) which carries it out of the system cycle. In this portion of the cycle the amount of heat rejected is Q_1. Thus the system functions to receive heat at a low temperature and pressure level and to discharge

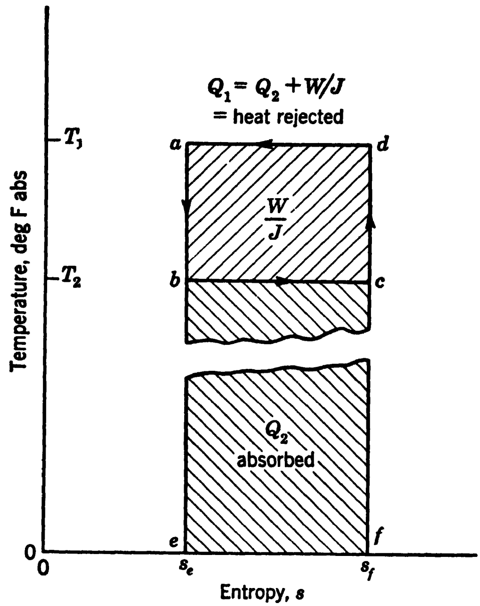


FIG. 285. Reversed Carnot cycle.

it at a higher pressure and temperature condition. Area *a–d–f–e–a* is proportional to the heat Q_1 and is the sum of Q_2 and W/J. In the Carnot cycle Q_1 is heat supplied for the production of power, and, in the reversed or refrigeration cycle, Q_1 is heat rejected.

296. Coefficient of Performance. The effectiveness of the reversed ideal cycle of a heat engine in producing refrigeration is designated by a term known as *coefficient of performance* (*cop*). The coefficient is the ratio of the refrigeration effect Q_2 to the work W/J necessary to produce it, both being expressed in Btu per pound of refrigerant. The item W/J is equal to $Q_1 - Q_2$. Also Q_1 is equal to $T_1(s_f - s_e)$, and Q_2 is equal to $T_2(s_f - s_e)$. Therefore, the coefficient of ideal performance may be expressed by one of the three following equalities.

$$cop_{ideal} = \frac{Q_2}{W/J} = \frac{Q_2}{Q_1 - Q_2} = \frac{T_2}{T_1 - T_2} \tag{319}$$

The required work decreases as $T_1 - T_2$ becomes smaller.

297. Units of Refrigeration Capacity. The latent heat of fusion of ice is a little less than 144 Btu per lb, but for the purpose of establishing refrigeration capacity the numerical quantity 144 is used. When ice was used as the principal means of cooling materials, a ton of refrigeration was defined as an equivalent to the amount of heat absorbed when 1 ton of ice, at 32 F, was converted into water at 32 F. This quantity of heat amounted to $2000 \times 144 = 288{,}000$ Btu with no specification as to the time required for the process to take place. Thus the time interval might be short or long, and in either case the ton of refrigeration would be the same. Therefore, a better and a common method is to define *a ton of refrigeration as the absorption of* 288,000 *Btu per* 24 *hr, or* 12,000 *Btu per hr, or* 200 *Btu per min.*, whch establishes the preceding quantities as rates.

Horsepower is expressed as a rate which is the performance each minute of 33,000 ft-lb of work. In terms of heat units 1 hp is $33{,}000/778 = 42.4$ Btu per min. Therefore, refrigerating machine performance may be expressed as horsepower per ton of capacity, which is

$$\text{hp per ton} = \frac{200}{cop\ (42.4)} = \frac{4.71}{cop} \tag{320}$$

When use is made of equation 320 an indication should be made relative to the coefficient of performance, *cop*, used; i.e., whether it is for the reversed Carnot cycle, Art. 295, or whether of an actual cycle as discussed in Art. 306.

298. Choice of a Refrigerant. The selection of a working medium or refrigerant, for a given type of cooling plant, is based on the pressures and temperatures at which it will vaporize, along with other essential factors. The latter include high latent heat of vaporization; odor; specific volumes; freedom from corrosion; possibilities of explosion and combustion; freezing temperature; stability; solvent and nonsolvent properties; reactions with lubricating oils; heat conductivity; high electrical resistance; noninjury to foods; toxic effects; and cost.

299. Refrigerating Media. Possible refrigerants are: air; ammonia, NH_3; carbon dioxide, CO_2; methyl chloride, CH_3Cl; sulfur dioxide, SO_2; water vapor, H_2O; methylene chloride, CH_2Cl_2, or Carrene No. 1; monofluorotrichloromethane, F-11, or Carrene No. 2; dichlorodifluoromethane, CCl_2F_2, or Freon 12 (F-12); and lithium bromide, LiBr, in solution with water. Properties pertinent to a limited number of refrigerants are listed in Table XII.

In the ordinary processes of refrigeration *dense air* is always in a gaseous state and is never reduced to its liquid phase. Because of the high operating pressures necessary and the power requirements, air has little usage as a refrigerant. *Carbon dioxide*, CO_2, is a safe refrigerant which

TABLE XII

Characteristics of Refrigerants*

Based on American Society of Refrigerating Engineers (ASRE) Rating

Property	Freon 12 (F-12), CCl_2F_2	Carbon Dioxide, CO_2	Ammonia, NH_3	Sulfur Dioxide, SO_2	Methyl Chloride, CH_3Cl	Carrene, CH_2Cl_2	Freon 22 (F-22), $CHClF_2$
Melting point, F	−252.0	−160.6	−107.9	−98.9	− 143.7	−142.0	−256.0
Boiling point, F	−21.6	−108.4	−28.0	14.0	−10.6	103.7	−41.44
Critical temp, F	233. 0	87.8	271.2	314.8	289.6	421.0	204.8
Critical press., psia	582.0	1071.0	1651.0	1141.5	969.2	670.0	716.0
Suction temp., F (including 9 F superheat)	14.0	14.0	14.0	14.0	14.0	14.0	14.0
Discharge temp, F	86.0	86.0	86.0	86.0	86.0	86.0	86.0
Liquid temp, F	77.0	77.0	77.0	77.0	77.0	77.0	77.0
Suction press., psia, 5 F	26.51	331.9	34.27	11.81	20.9	2.5″Hg	43.12
Discharge press., psia, 86 F	107.9	1043.0	169.2	66.5	96.0	21.6″Hg	174.4
Enthalpy of liquid, Btu per lb, at 77 F	25.56	70.50	128.50	39.01	43.30	26.2	33.11
Enthalpy at vapor, Btu per lb, 14 F	79.82	142.35	618.70	185.17	197.40	167.2	106.24
Entropy	0.17317	1.3085	1.3368	0.4002	0.4258	0.3618	0.2309
Specific gravity at +5 F	1.445	0.985	0.660	1.475	0.980	1.336	1.337
Temp after compression, F	106.5	176.1	222.7	202.9	165.5	....	141.5
Superheat at discharge, F	20.5	90.1	136.7	116.9	79.5	....	55.5
Refrigerating effect, Btu per lb	54.3	71.9	490.2	146.2	154.1	141.0	73.13
Specific volume at suction, cu ft per lb	1.516	0.293	8.377	6.880	4.656	48.31	1.275
Pounds of refrigerant per ton per minute	3.671	2.783	0.408	1.370	1.298	1.420	2.735
Cubic inches displacement, per ton per minute	9618	1409	5906	16,280	10,440	118,678.	6345
Refrigeration per cu ft, Btu	36.0	245.0	58.4	21.0	33.0	2.9	54.5
$k = c_p/c_v$	1.14	1.31	1.3	1.25	1.20	1.21	....
Compression ratio	4.07	3.14	4.93	5.63	4.58	8.50	4.04

* "Lubricants for Refrigeration Systems," The Texas Company. Used by permission.

can be liquefied more readily than air but its operating pressures are high and the power required per ton of refrigeration has restricted its use except in special cases.

Ammonia (anhydrous), one of the early refrigerants, has found a wide usage in ice-making and cold-storage plants. Ammonia is much less expensive than some refrigerants, is suitable for low-temperature operations, and requires a low volume of gas to be circulated per ton of refrigeration. Undesirable qualities of ammonia are its toxicity; bad effects on human eyes, skin, and lungs; possible explosive tendencies; disagreeable odor; and corrosion of copper, zinc, and brass.

At atmospheric pressure *methyl chloride* is a gas possessing physical and thermodynamic properties which make it a satisfactory refrigerant for small cooling and freezing units. In the presence of moisture it is corrosive to the point of destruction of zinc and aluminum equipment parts, and it produces solvent action on organic compounds and rubber. Since oil and methyl chloride mix together, provision has to be made for their separation when the two come into contact in machine lubrication. Methyl chloride is very stable, has a sweet odor, and when well diluted with air has little effect on the human body; heavy concentrations of the material in air may produce unconsciousness or even death.

Sulfur dioxide, when hermetically sealed within domestic refrigerating units, does not present a serious problem even though its toxic qualities are a hazard to be considered. It has a high condensing temperature, an evaporating temperature of about 14 F, and operates efficiently at low pressures. The material is stable, has a pungent odor, is irritating to the human body under most conditions, is not a contaminant of foods, and when mixed with water is a very corrosive agent.

Water vapor is inexpensive, nontoxic to the human body, easily controlled, and has the highest latent heat of vaporization of any refrigerant listed. Large volumes of vapor must be handled at low pressures when it is used, and the system must be free from air leaks; also it cannot be used to produce temperatures much below 40 F.

Various halogenated hydrocarbons compose the *Freon* group. They are especially desirable from the standpoints of densities, together with their freezing, boiling, and critical points. The Freons do not corrode metals, irritate the human body, produce objectionable odors, affect food products, or react with the lubricating oils that may be mixed with them. Care must be taken in Freon-filled systems to protect rubber, varnish, some insulating materials, and other substances from their solvent action.

Figure 286 is a pressure-enthalpy diagram which gives properties of

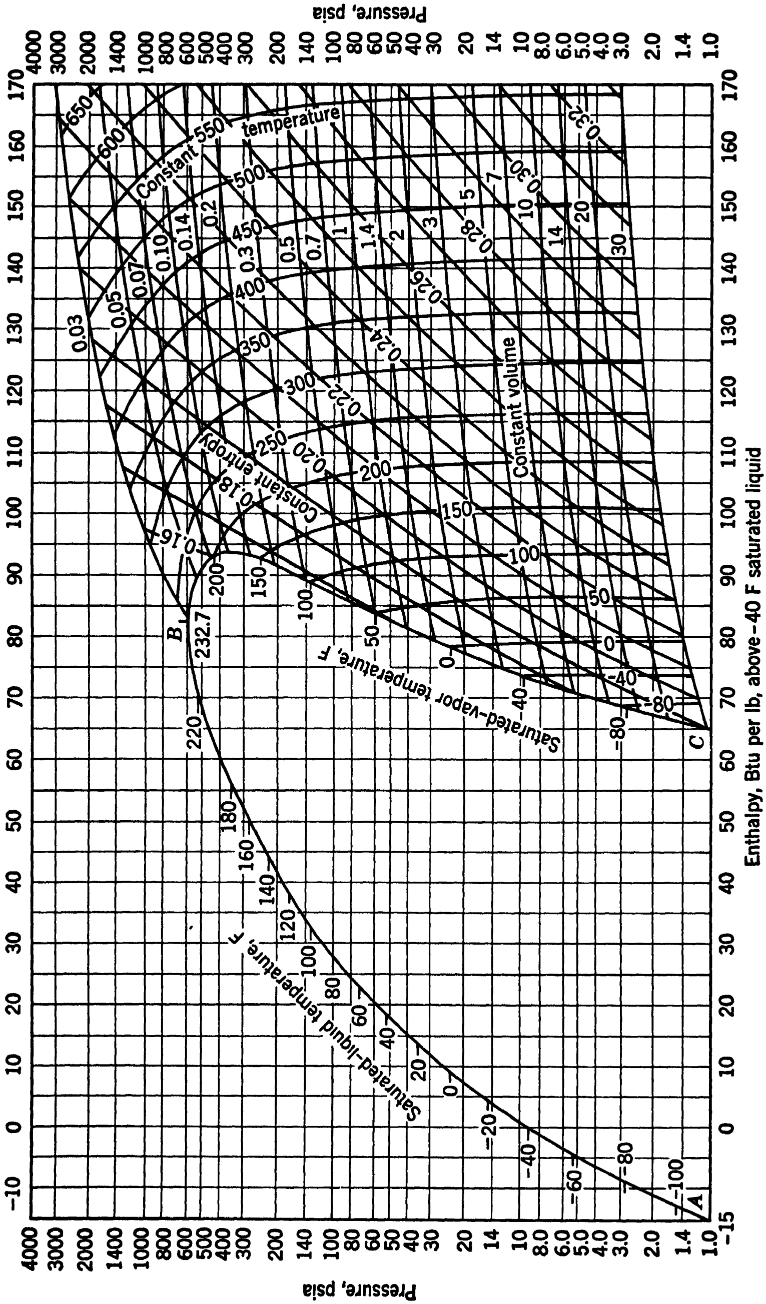


FIG. 286 Pressure-enthalpy diagram for dichlorodifluoromethane, F-12.

Freon 12 (F-12). Table XX (Appendix) presents in tabular form properties of F-12 which is dichlorodifluoromethane. Refrigerant pressure-enthalpy diagrams vary somewhat with the material involved.

300. Systems of Refrigeration. The systems of mechanical refrigeration fall under two general classifications: *compression* and *absorption.* Compression systems include water-vapor apparatus using steam jets or centrifugal compressors as well as piston and rotary compressors functioning with any suitable refrigerant, examples of which are listed in Art. 299.

An absorption system does not make use of a compressor but in its place utilizes a heater, a rectifier, a liquid cooler, and a pump to function with a suitable refrigerant which at times is held in a solution. The most commonly used solution is ammonia in water but more recently water[1] with dissolved lithium bromide, and methylene chloride in solution with either dimethyl ether or tetraethylene glycol have been used in absorption systems installed for the purpose of reducing the temperature of water used in air-cooling and dehumidifying plants.

301. Water-Vapor Refrigeration. Either steam jets or centrifugal compressors are used to produce a condition of a very low absolute pressure in a flash chamber and storage space where water is to be cooled. Relatively warm water entering the evaporating chamber is broken into fine droplets by spray nozzles, and as a result of a transformation of sensible to latent heat a part of it is flashed into water vapor. The action produces a lowering of the temperature of the unvaporized water. The vacuum required for the vaporization of the water ranges from 29.5 to 29.75 in. of mercury referred to a 30-in. barometer and is dependent upon the required final water temperature. The vacuum-producing equipment must remove from the evaporating chamber large volumes of water vapor together with entrained air and other gases. The vapor-gas mixture must be put under sufficient pressure to give rapid flow into a condenser. The build-up of pressure even though it is subatmospheric places the water-vapor equipment under the classification of a compression unit. The weight of water flashed into vapor amounts to about 0.1 per cent per deg F of cooling effected for each unit of water cooled.

302. Steam-Jet Refrigeration Apparatus. Commercial equipment for the production of water-vapor refrigeration by steam jets is alike in principle but varies somewhat in the details of construction. Figure 287 is a diagrammatic sketch of a steam-jet vacuum refrigerating unit fitted with a surface-type condenser. This type of apparatus can

[1] "New Absorption Refrigeration Unit Uses Water Vapor," A. A. Berestneff, *Power Engineering,* May, 1950, pp. 108.

be used with either barometric or low-level jet condensers. The motive steam supplied to the booster ejector may have a pressure as low as 2 to 3 psig, but when such a low pressure is employed the two-stage air ejector serving the booster ejector must function with steam at a reasonably high pressure. The cooled water of the apparatus shown by Fig. 287 flows in a closed circuit; i.e., water warmed in heat-exchanger equipment is returned to the flash chamber and is recirculated. Owing

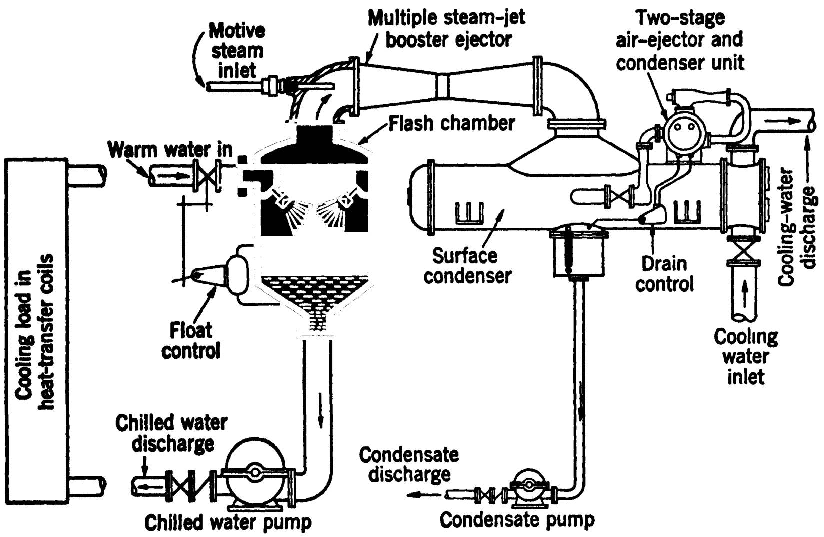


FIG. 287. C. H. Wheeler steam-jet vacuum refrigeration unit with surface condensers.

to the removal of vapor formed, some replenishment of water passing into the flash chamber must be made.

Whenever necessary two or more booster ejectors may be required to give the desired capacity. Either manual or automatic control of a booster ejector to give on and off operation may regulate the cooled-water temperature and the capacity produced. Where multiple boosters are used only a single booster is controlled automatically and the others are regulated manually, irrespective of any arrangement for their operation in parallel. The only rotating parts in the steam-jet unit illustrated are the impellers of the two centrifugal water-handling pumps.

303. Steam Requirements for Jet Refrigeration. Steam-jet refrigerating equipment is not well adapted to units of less than 6 tons capacity. Considerable condenser cooling water is necessary, the amount

per ton being dependent upon the size of the machine and the initial temperature of the water as it enters the condenser. The steam consumption varies with the initial steam pressure and the vacuum maintained in the flash chamber.

Steam consumptions[2] per ton-hour have been reported for the following steam pressures: 10 psig, 50 to 60 lb; 100 psig, 30 to 35 lb; and 200 psig, 20 to 25 lb. The foregoing are for the conditions of 85 F entering condenser water, chilled water leaving at 45 F, and 6 to 8 gpm of condensing water per ton of refrigeration.

304. Steam-Jet Refrigeration Capacity. When there is no flow of heat through the walls of the flash chamber certain equalities may be established. These include the weight of water entering the flash chamber and the sum of the chilled-water weight and that of the vapor withdrawn by the ejector. The entering weight of warm water in pounds per unit of time is designated by m_1 and is the sum of the weight of the chilled water m_2 discharged per unit of time and m_3 the weight of water flashed into vapor during the same time interval. The enthalpies of the liquid for the warm- and the chilled-water conditions are $h_{f,1}$ and $h_{f,2}$, respectively. The total enthalpy of the wet saturated vapor $h_{gw,3}$ is dependent upon its quality and either its temperature or absolute pressure. All enthalpies are in Btu per pound.

$$m_1 = m_2 + m_3$$

$$m_1 h_{f,1} = m_2 h_{f,2} + m_3 h_{gw,3}$$

$$m_2(h_{f,1} - h_{f,2}) = m_3(h_{gw,3} - h_{f,1}) \qquad (321)$$

Either quantity of equation 321 represents the heat taken from the water to produce the temperature of the chilled fluid. The capacity of the machine, when m_2 represents pounds per minute, is

$$\textit{Capacity in tons} = m_2(h_{f,1} - h_{f,2})/200 \qquad (322)$$

Example. A manufacturer of steam-jet refrigeration equipment indicates the following data of operation for a unit of apparatus: barometer, 29.921 in. of mercury; vacuum in flash chamber, 29.673 in. of mercury; and vapor temperature in surface condenser, 100 F. The water passing through the flash chamber enters at 50 F and leaves at 40 F. The discharge from the flash chamber is 500 gpm of water at 40 F and vapor with a quality of 0.99. Find (*a*) the refrigeration capacity in tons; (*b*) the weight of vapor passing to the ejector, pounds per minute; (*c*) the ratio of absolute pressure in the condenser to that in the flash chamber; and (*d*) the volume of vapor compressed.

[2] "New Absorption Refrigeration Unit Uses Water Vapor," A. A. Berestneff, *Power Engineering*, May, 1950, pp. 108.

Solution. (*a*) Weight of chilled water per minute,

$$m_2 = 500 \times 62.42 \times 231/1728 = 4172.4 \text{ lb}$$

$$\text{Tons capacity} = 4172.4(18.07 - 8.05)/200 = 209$$

(*b*) $h_{gw,3} = 8.05 + (0.99 \times 1071.3) = 1068.6$ Btu per lb

$$m_3 = 4172.4(18.07 - 8.05)/(1068.6 - 18.07) = 39.7 \text{ lb per min}$$

(*c*) Pressure ratio developed $= \dfrac{29.921 - 27.989}{29.921 - 29.673} = \dfrac{1.932}{0.248} = 7.8$ which indicates the necessary performance of the booster ejector as a compressor.

(*d*) Total volume of vapor handled at 40 F is

$$39.7[0.016 + (0.99 \times 2444)] = 39.7[0.016 + 2419.6] = 39.7 \times 2419.61 = 95{,}960 \text{ cfm}$$

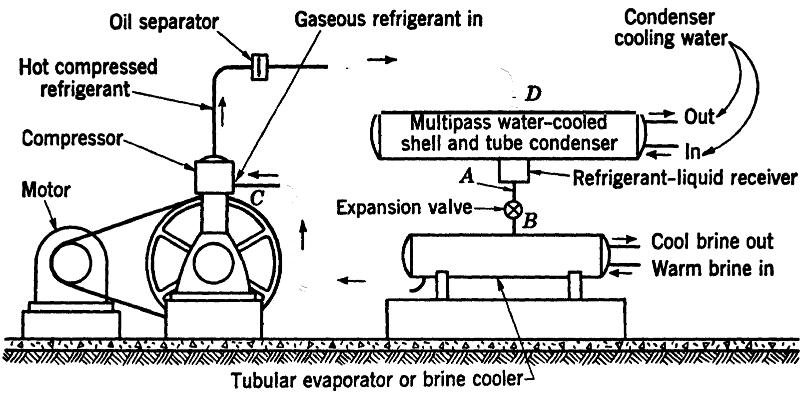


FIG. 288. Diagrammatic layout of a mechanical-refrigeration plant with a reciprocating-piston compressor.

305. Refrigeration with a Reciprocating Compressor. Much of the material of this article is applicable to installations in which pressure is built up within a suitable refrigerant by means of a centrifugal compressor. The order of events in a mechanical refrigeration plant with a reciprocating-piston compressor is as indicated by Fig. 288, in which a single-cylinder single-acting unit is shown. The refrigerant, as a gas, enters the compressor cylinder at *C*, and its pressure is increased by polytropic compression after which the medium flows to some suitable condenser described in Art. 312. In the condenser the refrigerant, except in special cases, is reduced to a liquid under pressure, and a large part of its heat is carried away by the condenser coolant (either air or water). The liquefying process in the condenser is not an isothermal one, as the temperature differential maintained at different locations varies. The condenser is the piece of equipment in which

heat is discharged from the system at a high-temperature level. The *high-pressure side* of the system extends through *C–D–A* to the expansion valve, which is either a hand- or a temperature-regulated control. The expansion valve regulates the rate of flow of the refrigerant into the low-pressure low-temperature evaporator or cooler, where it vaporizes. During this action sensible heat of the refrigerant and that absorbed in the evaporator produce refrigerant vapor between the expansion valve and point *C*. Usually the expansion valve is located close to the evaporator refrigerant inlet at *B*. Thus the low-pressure side of the system is from *B* to *C*.

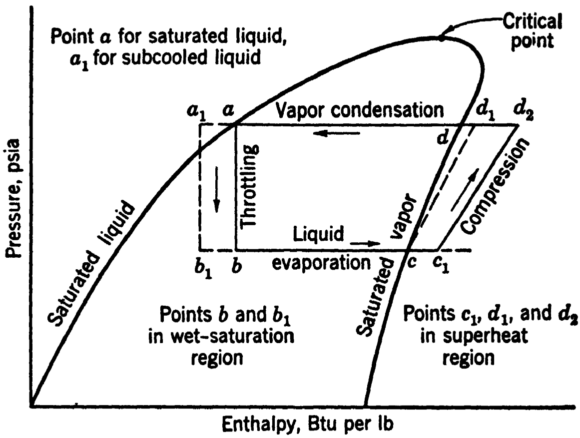


FIG. 289. Pressure-enthalpy diagrams for simple refrigeration cycles.

306. Commercial Refrigeration Cycle. A commercial refrigerating machine cannot fulfill the requirements of the theoretical cycle. In an actual process the liquid refrigerant arrives at the inlet of the expansion valve in either a saturated or a subcooled state. Inasmuch as isentropic expansion is not feasible, the action of the expansion valve is to throttle the refrigerant (in what amounts to an orifice with a variable opening) to produce a constant-enthalpy condition, as shown by the line between *a* and *b* on Figs. 289 and 290. The throttling process with constant enthalpy produces an increase of the entropy of the refrigerant as its temperature and pressure decrease to meet the necessary requirements in the evaporator. For the solution of practical problems Figs. 286 and 289 are more useful than Figs. 285 and 290.

In the actual process the temperature of the boiling refrigerant in the evaporator must fall below that of the medium to be cooled. The pressure maintained in the evaporator is fixed by the refrigerant temperature

necessary to give the desired cooling. In an actual cycle the refrigerant vapor may leave the evaporator in either a wet-saturated, a dry-saturated, or a superheated condition. Generally some superheat as shown by point c_1 of Fig. 289 is desirable. Polytropic compression occurs in the actual cycle, and the terminal condition of the compressed refrigerant is partially dependent upon its initial state. Superheat at the end of compression (point d_1 or d_2, Fig. 289) is desirable. The heated medium, in its discharge condition, flows to the condenser where

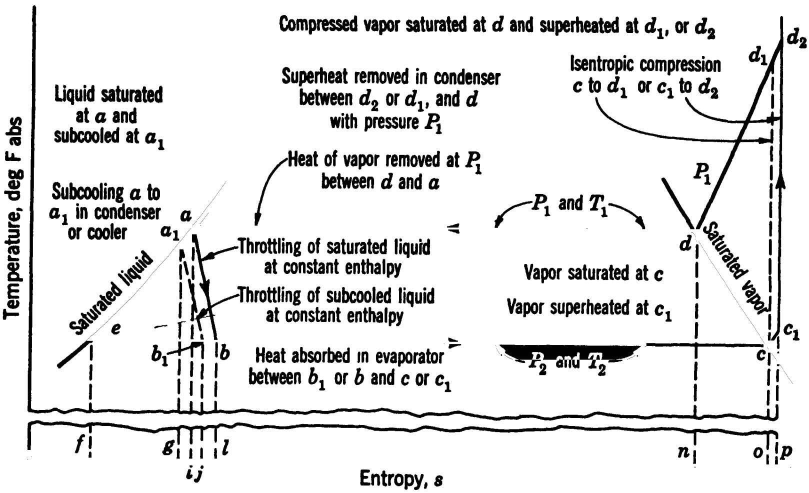


FIG. 290. Temperature-entropy diagram for a refrigerant with or without subcooling before entering the evaporator and with saturated or superheated vapor leaving the evaporator.

the coolant is maintained at a temperature below that of the refrigerant. In the condenser the refrigerant loses enthalpy of superheat, enthalpy of evaporation or latent heat of vaporization, and if the liquid is subcooled some enthalpy of the liquid. In each step of this actual cycle (not that of the compressor alone) the processes are irreversible.

307. Modifications of Compression Refrigeration. More complex systems of compression refrigeration than the simple one of Art. 305 are in use. The compression of gaseous vapor may be accomplished by the use of a two-stage compressor with an intercooler placed between the low- and the high-pressure stages to function much the same as two-stage air compressors described in Art. 255. Quite often because of the relatively warm cooling water available the use of two-stage compression with an intercooler is not feasible. Mention was made of the flashing of refrigerant liquid into vapor just after its passage through

the expansion valve. The flashed vapor is of no value for the absorption of heat in the evaporator but it does impose an additional load on the refrigerant compressor. Remedial action can be secured by the use of a two-stage compressor and two expansion valves with a chamber for the collection of the flashed vapor placed between the two valves. The flash vapor from the first expansion valve is withdrawn from the flash chamber and is drawn into the second stage of the refrigerant compressor. In this way the withdrawn vapor does not pass through the evaporator. Also the vapor is withdrawn at an intermediate pressure, and less work of compression is required. When refrigeration cooling is required at two different temperature levels a dual system may be used with two different refrigerants in simple cycles which are operated in series. In this arrangement, sometimes called a *cascade system*, the condenser of the low-temperature cycle is utilized as the evaporator for the high-temperature cycle.

308. Pressure-Enthalpy Relations in Refrigeration Cycles. The chief advantage of the pressure-enthalpy diagram lies in the fact that the energy changes representing either constant-enthalpy or constant-pressure processes are shown as straight lines. The skeleton pressure-enthalpy diagram of Fig. 289 is incomplete in the way of all factual data for the refrigerant which it represents; data not given may be found in complete tables. The graphical diagram is an aid in visualizing the refrigeration processes together with making calculations for them. More accurate results may come from the use of tabular data interpolated for both temperature and pressure conditions.

Two conditions of operation are shown by the pressure-enthalpy diagram of Fig. 289. Condition a–b–c–d_1–a represents throttling of saturated liquid from a to b, vaporization of the liquid refrigerant from b to c to give saturated vapor in a dry state at c, dry compression of the vapor from c to d_1 to produce some superheat at d_1, and the removal of superheat and enthalpy of evaporation to give saturated refrigerant liquid at a. When subcooling occurs in the condenser, the state point of the liquid is shown by a_1 and the refrigerant entering the evaporator by b_1. When the refrigerant leaves the evaporator in a superheated condition its compression is represented by line c_1d_2. These latter conditions of operation are an aid in increasing the amount of refrigeration produced per pound of refrigerant handled.

309. Enthalpy Changes in a Compression Cycle. Refrigeration compressors are often rated on a basis of the working medium entering their cylinders in a saturated condition at 5 F and being discharged to the condenser at 86 F. Other conditions mentioned involve having the refrigerant subcooled by 9 F as it reaches the expansion valve and hav-

ing the vapor from the evaporator superheated by 9 F as it reaches the compressor suction inlet.

In actual events the enthalpies in Btu per pound of refrigerant at the various points in a compression cycle are indicated by the symbol h which includes a proper subscript. When the refrigerant enters the compressor as a mixture of vapor and liquid (quality less than 1.0) the compression is wet, although the discharge condition may be wet or dry saturated or superheated vapor. If the refrigerant is dry and saturated or superheated at the beginning of compression, its terminal condition will be a superheated one and the compression is said to be dry. The enthalpies involved are dependent upon qualities, pressures, and temperatures, and tables of both saturated and superheated refrigerants as well as Mollier diagrams may be required to determine them.

The energy equations, per pound of refrigerant m, which follow apply to the conditions a_1–b_1–c_1–d_2–a_1 of Fig. 289; these with modifications may be applied when the state points are a–b–c–d_1–a. The enthalpy $h_{a,1}$ at a_1 is equal to the enthalpy $h_{b,1}$ at b_1. The refrigeration effect in the evaporator is

$$Q_2 = (h_{c,1} - h_{b,1}) \tag{323}$$

and the work of isentropic compression is

$$W_i = (h_{d,2} - h_{c,1}) \tag{324}$$

Subcooling of the liquid refrigerant is involved in this case, and the heat removed in the condenser Q_1 is

$$Q_1 = (h_{d,2} - h_{a,1}) \tag{325}$$

Equation 325 includes the removal of the enthalpies of superheat, saturated vapor, and liquid. Q_1 may also be written as $Q_2 + W_i$ when both items are expressed in heat units.

For a given refrigeration capacity, in tons, the flow of refrigerant, in pounds per minute, is

$$m_t = \frac{200 \times \text{tons}}{Q_2} \tag{326}$$

The isentropic horsepower requirements of a compressor may be estimated by any of the following equations

$$hp_i = \frac{m_t \times W_i \times 778}{33{,}000} = \frac{m_t \times W_i}{42.42} \tag{327}$$

$$= \frac{200 \times \text{tons} \times W_i}{Q_2 \times 42.42} = \frac{4.717 \times \text{tons} \times W_i}{Q_2}$$

$$= \frac{200 \times \text{tons} \times (h_{d,2} - h_{c,1})}{Q_2 \times 42.42} \tag{328}$$

The isentropic work of compression, when $k = c_p/c_v$, is

$$W_i = \left(\frac{k}{k-1}\right)\left(\frac{P_c v_c}{778}\right)\left[\left(\frac{P_d}{P_c}\right)^{(k-1)/k} - 1\right] \tag{329}$$

where W_i = isentropic work done in compressor, Btu per lb.
P_c = pressure of refrigerant entering compressor, psfa.
P_d = pressure of compressor discharge, psfa.
v_c = volume of refrigerant entering compressor, cu ft per lb.

The work as given by equation 324 is positive, and so equation 329 is arranged to give the same sign. Values of c_p and c_v vary with different pressures and temperatures for a given refrigerant and also with the different media used in compressors.

The actual brake horsepower necessary is generally 25 to 30 per cent greater than the isentropic horsepower. Therefore, the ratio, n_e, of ideal power requirements to those actually necessary ranges from 0.75 to 0.80. The brake horsepower, based on isentropic compression between c_1 and d_2, Fig. 289, is

$$\text{bhp} = \frac{4.717(h_{d,2} - h_{c,1}) \text{ tons of refrigeration}}{n_e \times Q_2} \tag{330}$$

The actual coefficient of performance for a compression refrigeration system is

$$cop_{actual} = \frac{\text{Refrigeration effect}}{\text{Actual work input}} \tag{331}$$

When all items are based on either brake horsepower per ton of refrigeration or per pound of refrigerant,

$$cop_{actual} = \frac{200}{42.42 \text{ bhp}} = \frac{4.717}{\text{bhp}} \tag{332}$$

Example. From data given in Table XII calculate for ammonia the following items: (*a*) the value of $k = c_p/c_v$; (*b*) the degrees of superheat in the ammonia as discharged to the condenser; (*c*) the refrigeration effect, Btu per pound; (*d*) weight of refrigerant, pounds per ton per minute; (*e*) cubic inches displacement, per ton per minute; (*f*) refrigeration per cubic foot, Btu; and (*g*) compression ratio. Parts of the data of Table XII are taken from tables giving the properties of ammonia as a saturated liquid, saturated vapor, and in a superheated condition.

Solution. (*a*) $T_{d,2} = T_{c,1}\left(\frac{P_d}{P_c}\right)^{(k-1)/k}$ or $T_{d,2} = T_{c,1}\left(\frac{144 \times 169.2}{144 \times 34.27}\right)^{(k-1)/k}$

$$(460 + 222.7) = (460 + 14)(4.93)^{(k-1)/k}$$

$$682.7 \div 474 = 4.93^{(k-1)/k} \quad \text{or} \quad 1.44 = 4.93^{(k-1)/k}$$

$$\log 1.44 = \log 4.93 \left(\frac{k-1}{k}\right) = 0.1584 = 0.6928 \left(\frac{k-1}{k}\right)$$

$$\frac{k-1}{k} = 0.229 \quad \text{or} \quad 0.229k = k - 1$$

$$0.771k = 1 \quad k = 1.295$$

(*b*) Superheat $= 222.7 - 86 = 136.7$ F
(*c*) Refrigeration effect $= 618.8 - 128.5 = 490.2$ Btu per lb
(*d*) Weight of refrigerant per ton per min $= 200/(618.7 - 128.5) = 0.408$ lb
(*e*) Volume of piston displacement per ton per min $= 0.408 \times 8.377 \times 1728 = 5906$ cu in.
(*f*) Refrigeration per cu ft of refrigerant $= 490.2 \div 8.377 = 58.4$ Btu
(*g*) Compression ratio $= 169.2 \div 34.27 = 4.93$

Example. A motor-driven single-cylinder compressor functions in an ammonia system to give a refrigeration effect of 15 tons when it receives refrigerant at 34.27 psia and 14 F. The pressure at the end of compression is 169.2 psia when the condenser temperature is 86 F and the ammonia-liquid temperature is 77 F. Find (*a*) the weight of refrigerant supplied to the evaporator, pounds per minute; (*b*) both the work of isentropic compression, when $k = 1.30$, and the work of compression based on the enthalpy change in the compressor measured in Btu per minute; (*c*) weight, pounds per minute, of condenser cooling water when its temperature rise is 15 F; (*d*) the isentropic horsepower, hp_i, produced in the compressor cylinder; (*e*) the brake horsepower of the compressor if n_e is 0.78; and (*f*) the actual coefficient of performance. The enthalpy at the end of compression (from a *P-h* diagram for ammonia) is 720.7 Btu per lb. Refer to Table XII for enthalpies not given.

Solution. (*a*) $m_t = (200 \times 15) \div (618.7 - 128.5) = 3000 \div 490.2 = 6.12$ lb

(*b*) Work of isentropic compression

$$= W_i = \left(\frac{1.30}{1.30-1}\right)\left(\frac{144 \times 34.27 \times 8.377}{778}\right) \times \left[\left(\frac{144 \times 169.2}{144 \times 34.27}\right)^{(1.3-1)/1.3} - 1\right] 6.12$$

$$= 4.33 \times 53.1 \times 0.443 \times 6.12 = 623 \text{ Btu}$$

Also $W_i = 6.12(720.7 - 618.7) = 624$ Btu

(*c*) $m_w = 6.12(720.7 - 128.5) \div 15 = 241.7$ lb

(*d*) $hp_i = 623 \div 42.42 = 14.71$

(*e*) $$bhp = \frac{4.717(720.7 - 618.7)15}{0.78 \times 490.2} = 18.85$$

(*f*) $$cop_{actual} = \frac{4.717 \times 15}{18.85} = 3.75$$

310. Compressors. Machines with moving parts, used for the reduction of a refrigerant volume together with an increase of its pressure, fall into the following general types: reciprocating, rotary, and

centrifugal compressors. Compressors must be adapted to the refrigerant used with them; i.e., ammonia cannot be used in a compressor designed for CO_2 or F-12.

The classifications of reciprocating compressors are based on: the number of cylinders; the cylinder position, horizontal, or V-placement, or vertical; the piston action, single-acting or double-acting; piston

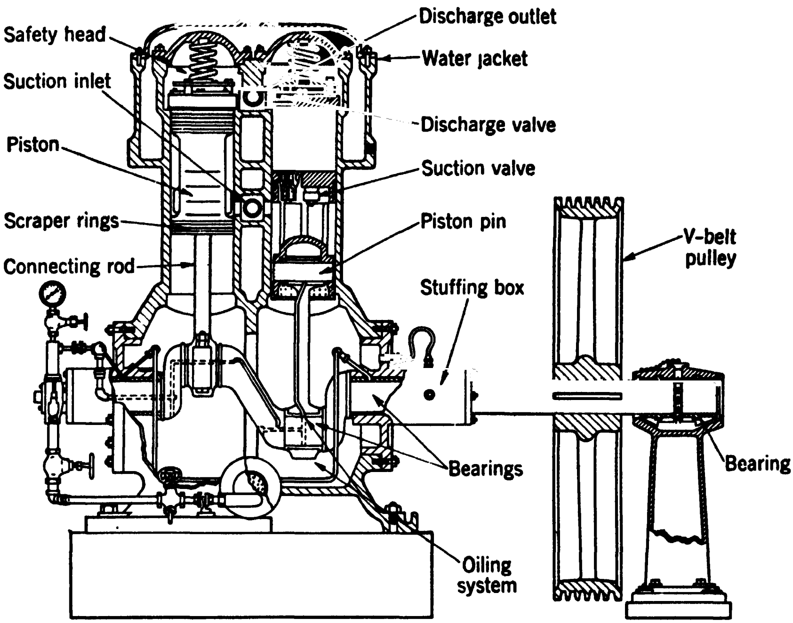


FIG. 291. Frick vertical two-cylinder single-acting belt-driven Freon 12 compressor.

construction, trunk or units with piston rods and crossheads; the number of stages of compression, single-stage, or two-stage; drive arrangements, engine or motor with either belt or direct connections; and the placement of their inlet and outlet valves. Small piston compressors usually have vertical placements of one or more cylinders, as is typified by the two-cylinder single-acting belt-driven unit of Fig. 291. For large industrial installations horizontal double-acting units providing two-stage compression in two or more cylinders may be used. Reciprocating compressors are not adapted to high rotative speeds, they occupy considerable floor space per unit of capacity, and they are subject to vibrations. However, piston units do give a better volumetric efficiency than is the case with rotary units.

Rotary compressors can be built as sealed units, which is an advan-

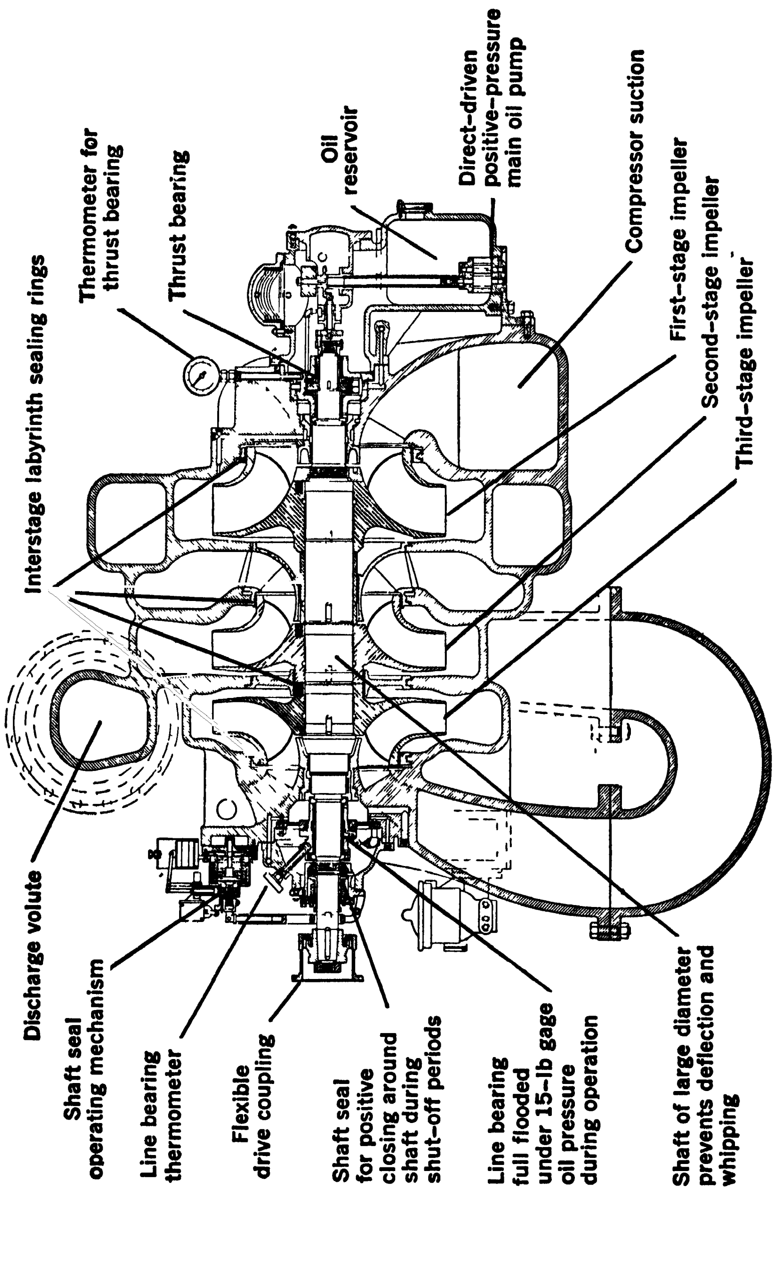


FIG. 292. Sectional elevation of a Worthington three-stage volute centrifugal refrigeration compressor.

tage in maintaining freedom from refrigerant leaks. Generally such units have some sort of rotating part mounted eccentrically upon a shaft within the casing. Other forms of construction include those having vanes which slide in radial slots of the rotor about which the casing is placed eccentrically. In general these rotating parts entrap gaseous refrigerants ahead of them and build up its pressure in much the same fashion as is true of rotary pumps (Art. 244) handling liquids.

Centrifugal compressors operate at high speeds and consist of a shaft with blades mounted upon it which is enclosed by a housing with a scroll of the order of that of a centrifugal pump. The rotor shaft is supported at each end by suitable bearings, one of which is capable of absorbing the end thrust upon the shaft. Figure 292 illustrates a three-stage centrifugal compressor. The impellers of the first and second stages produce a thrust toward the right or the suction inlet end of the unit. The third-stage impeller delivers refrigerant at its highest pressure and is so mounted on the shaft that its thrust opposes the resultant thrust of the first two stages. Consequently the simple thrust bearing at the suction end of the shaft is ordinarily able to handle any unbalanced forces.

Discussions have been given in prior articles dealing with thermodynamics, air compression, and internal-combustion engines relative to the compression of gases and air. The material dealing with the use of piston compressors with air applies equally to piston refrigeration compressors, provided the proper data are used, with respect to the estimated conditions at different locations within a cylinder and the effects of volumetric clearance upon its performance.

311. The Absorption Refrigeration System. The flow diagram of a simple absorption system is shown in Fig. 293. An absorber, a generator, and a pump replace the compressor of a compression system. The function of the three pieces of equipment is to increase the pressure of the refrigerant to that desired in the condenser. Otherwise the two systems are identical. The low-pressure refrigerant vapor leaving the evaporator passes to the absorber. Here it is taken up by a weak solution of the refrigerant dissolved in a solvent. Ammonia, NH_3, in solution with water has a wide usage in absorption systems. The strong solution formed within the absorber is pumped to the generator, where, by the addition of heat energy, refrigerant vapor is liberated and a satisfactory condenser pressure is produced. The resulting weak solution is returned from the generator to the absorber. More pieces of equipment are required for absorption systems than for those using compressors. An energy balance may be written for the complete system as well as for each component part.

On the market is an absorption system using water as a refrigerant with lithium bromide as the absorbent. Another unit employs methylene chloride as a refrigerant in solution with either dimethyl ether or tetraethylene glycol. These machines are used to cool water for use in summer air-conditioning plants, to cool spaces for the storage of foods, or to cool spaces where certain manufacturing operations are carried on. In general the solvents used with refrigerants range from

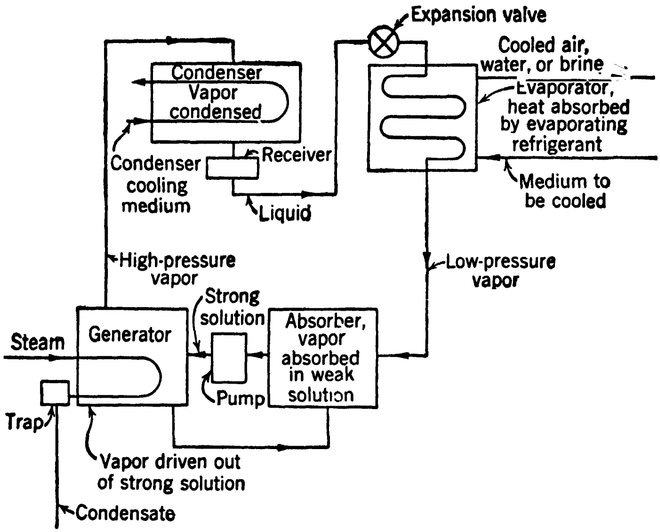


FIG. 293. Flow diagram for an absorption refrigeration plant.

liquids to solids, and the number of fluids required in an absorption may be one or several.

Except for small air-cooled household units, low-pressure heating steam and ample cooling water are required for the operation of absorption systems. Small refrigerators are made to function using heat energy obtained by the combustion of either gas or kerosene in a burner which is a part of the mechanism. Also in small refrigerators the circulation of the refrigerant and its solvent may be secured by the use of a bubble pump which functions in a manner similar to the tube of a coffee percolator.

312. Condensers. The disposal of heat picked up in a refrigerating system may be made by means of either an air-cooled or water-cooled condenser which liquefies the refrigerant. The air-cooled units are employed mostly in small household refrigerators. Water-cooled units may be one of the following forms of construction: (1) double-

tube devices; (2) one or more single large shells with one or more water passes; (3) multiple small shells each filled with water tubes and having counterflow of refrigerant and gases; (4) and evaporative condensers.

The condenser of Fig. 294 is of the double-tube type with water flowing through the inner tubes and with the gaseous refrigerant and its condensed liquid occupying the spaces between the inner and the outer tubes. This form of construction employs only horizontal tubes.

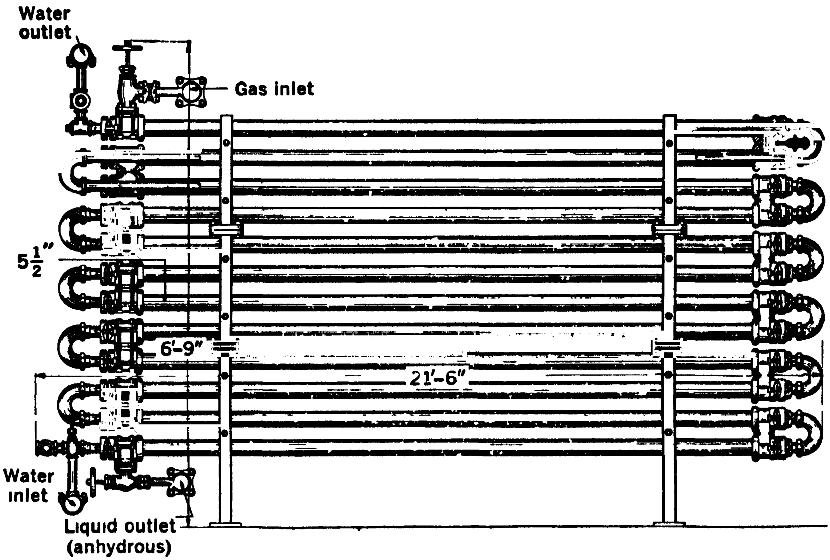


FIG 294 Vogt double-tube refrigeration condenser.

Large-shell condensers have many internal tubes through which water is circulated. For vertical installations the water makes only a single pass through the tubes, from the top to the bottom of the units. In horizontal placements the cooling water makes multiple passes through the tubes, owing to the action of baffles which are a part of the shell heads.

The condenser of Fig. 295 has multiple small shells fitted with water tubes which are surrounded by the refrigerant. The interconnections of the shells are made to promote counterflow of the refrigerant and the cooling water.

Evaporative condensers,[3] Fig. 296, have considerable usage where cooling water must be conserved. In operation the cooling water is sprayed upon the condenser tubes which are located within a housing. Air is drawn over the condensing coil by the action of a fan. Spray

[3] *From Bulletin D-350*, "Evaporative Condensers," The Trane Company.

water flows over the cooling surfaces in a direction counter to that of the air flow. Cooling of the spray water takes place as a part of it is evaporated when sensible heat is transformed to latent heat. The un-

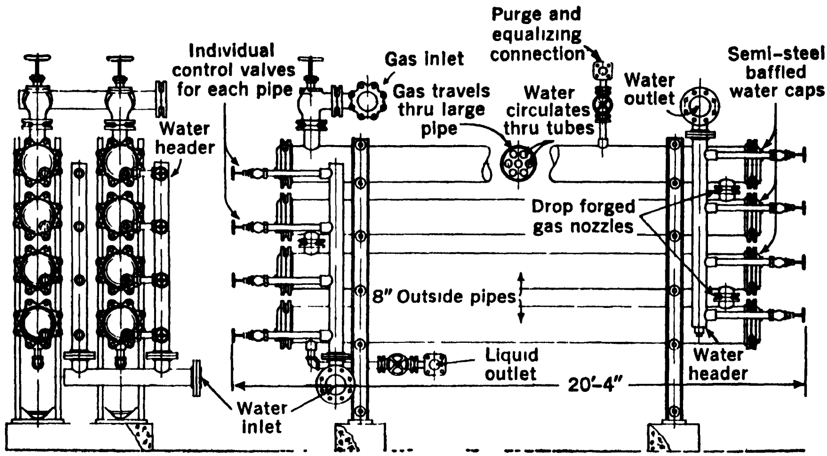


FIG. 295. Vogt multiple small shell multitube-type refrigeration condenser.

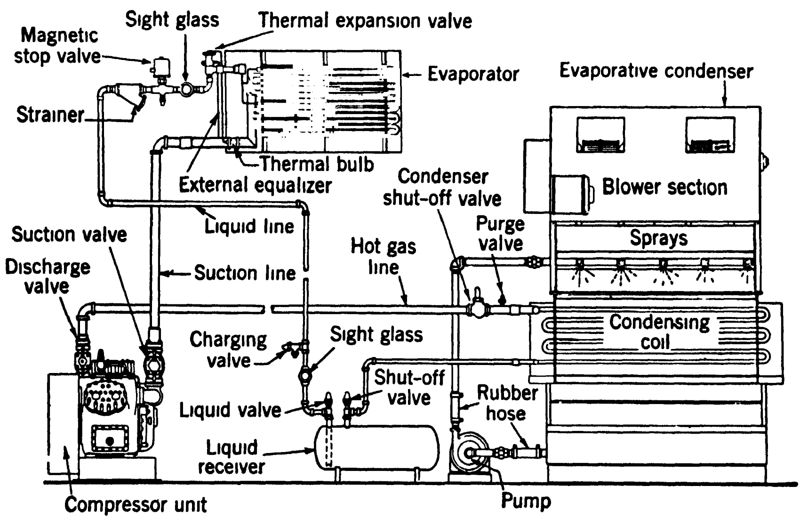


FIG 296. Compression refrigeration system with an evaporative condenser.

vaporized water collects in a sump at the bottom of the unit and is re-circulated by the spray pump. The loss of water due to evaporation amounts to about 10 per cent of that handled so that the saving of

water is approximately 90 per cent of the total requirements. Evaporative condensers may be located either outdoors or inside a building. When the interior location is used air supply and discharge ducts are required.

313. Evaporators. In a refrigeration system the refrigerant absorbs heat from the material to be cooled in apparatus used to lower the temperature of either water, brine, or air. The apparatus (evaporators) in which the cooling may be done is often classed as either a water, brine, or air cooler. Evaporators may be simply pipe coils or cast-iron sections, similar to those of heating radiators, mounted upon the walls of a space to be cooled. Other forms of construction embody the features of shell and tube condensers, double-tube condensers, or finned-tube air cooling coils as in Fig. 296. In any event proper distribution of the refrigerant within the evaporator is essential to its satisfactory performance.

314. Space Heating with a Refrigeration System. All-year air-conditioning systems may employ refrigeration machinery for cooling spaces in summer and heating them in winter. Two terms are applied to the process: heat pump and reversed-cycle refrigeration. All mechanical refrigeration equipment functions as a heat pump. The term reversed-cycle refrigeration is not a good one, as the equipment operates in a normal manner with heat discharged from a condenser at the high-temperature level. The refrigeration effect produced within an evaporator is employed for space cooling, and the heat discharged in a condenser is used for space warming.

A dual-purpose heat pump, adjusted for winter heating, is shown in Fig. 297. In this unit the flow of refrigerant is always in the same direction when the apparatus functions either in summer or winter. For space warming the equipment of Fig. 297 utilizes heat energy accruing from the input of power to the compressor and that obtained by cooling outside air in the evaporator section. Fan 1 draws outside air over the evaporator where vaporizing refrigerant cools it below its entering temperature. The warmed refrigerant vapor passes to the compressor, where its temperature and pressure are increased. Fan 2 moves recirculated air and possibly ventilating air over the condenser tubes, where the air in passing picks up heat and the refrigerant is reduced to a liquid state. Fan 2 delivers warmed air to the conditioned space, and the liquid refrigerant passes through the expansion valve to the evaporator to complete the refrigeration cycle.

When all dampers are turned through an angle of 90 deg the system, with normal functioning of the refrigeration equipment, delivers cooled air to the conditioned space. In summer cooling operation, the outside

air passes over the condenser to liquefy the refrigerant and the evaporator cools the recirculated air. When operated for summer cooling, fan 1 returns heated air to the outside and fan 2 forces cooled air to the conditioned space.

Sources of heat energy are: (1) cool or cold outside air, (2) water circulated through pipes buried in the earth, (3) water drawn from either

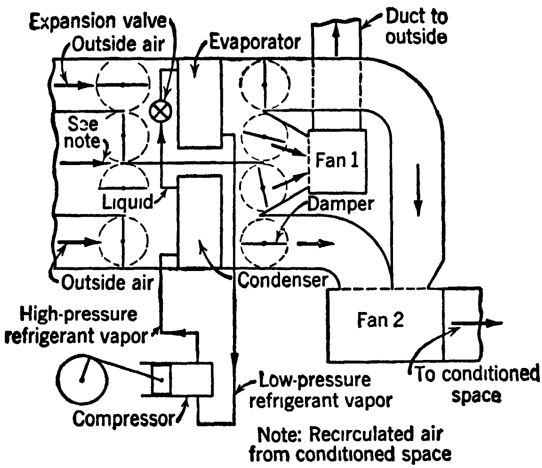


FIG. 297. Dual-purpose heat pump adjusted for winter heating.

lakes or rivers, (4) water from suitable wells, and (5) solar energy, which is little used. Of the possible supplies mentioned, well water is the most reliable and steady source of heat.

Depending upon the fluids cooled and heated in the evaporators and condensers, heat-pump operations may be classified as: (1) air to air with refrigerant flow reversed, (2) air to air with fixed refrigerant circuit, (3) liquid to air, reversed refrigerant flow, and (4) liquid to air with fixed refrigerant circuit.

One form of heat-pump design involves the operation of a system with reversed flow of the refrigerant flow. This action is secured by the use of 4-way valves in the refrigerant piping. The application of reversed refrigerant flow requires more component parts in a system but allows somewhat better control of the apparatus.

The selection, installation, and operation of suitable control equipment are necessary for the successful and satisfactory performance of the plant. The problems of heat distribution in and the matter of cooling conditioned spaces include the selection of a conveying medium and the maintenance of uniform and satisfactory air motion and tempera-

tures in occupied zones. Engineering problems include the cost of the installation and the noises produced by machinery.

PROBLEMS

1. (*a*) Eight-hundred gallons per minute of water measured at its inlet temperature of 50 F are cooled to 40 F in a refrigeration unit. (*a*) Find the tons of refrigeration effect. (*b*) Had the same quantity of brine with a specific gravity of 1.12 been used the enthalpies of the liquid, above 32 F, would have been 14.28 and 6.33 Btu per lb, respectively, at 50 F and 40 F. Find the tons of refrigeration produced under such conditions.

2. A refrigerant boils in an evaporator with an increase of enthalpy of 350 Btu per lb when 360 lb per hr are handled. Express the heat absorption in tons of refrigeration capacity. 10.5 tons, *Ans.*

3. In a reversed Carnot cycle the working medium absorbs heat at a temperature of -10 F and discharges it at 90 F. Find the theoretical coefficient of performance, the horsepower per ton of refrigeration, and the foot-pounds of work per pound of refrigerant. The initial and final enthalpies of the working medium are 0.296 and 1.356, respectively. 4.5, 1.05, and 82,468 ft-lb, *Ans.*

4. Dry saturated Freon 12 exists at a temperature of 90 F. (*a*) From the data of Table XX of the Appendix find all its properties. (*b*) Find the properties of Freon 12 at 114.3 psia and 50 F superheat.

5. Figure 286 indicates what condition for Freon 12 (*a*) to the left of the saturated-liquid curve (*b*) between the saturated-liquid line and the saturated-vapor curve and (*c*) to the right of the saturated-vapor curve?

6. A steam-jet vacuum refrigeration system is to produce 400 tons of refrigeration capacity. Operating conditions are to be: temperature of water entering flash chamber, 55 F; temperature of cooled water leaving flash chamber, 45 F; barometric pressure, 29.25 in. of mercury; vacuum in flash chamber, 28.95 in. of mercury; pressure in condenser, 1 psia; and flash-chamber vapor quality, 0.98. (*a*) Find the weight of chilled water discharged per minute. (*b*) Compute the weight of water vapor withdrawn per minute by the ejector. (*c*) Calculate the pressure ratio developed. (*d*) Find the total volume of vapor, cubic feet per minute, handled at 45 F and 0.98 quality. (*e*) Water at 60 F is mixed with return water supplied to the flash chamber. At what temperature is the return water if the temperature of the entering mixture is 55 F?

7. (*a*) What factors affect the steam consumption of steam-jet vacuum refrigeration units? (*b*) For such units, state the minimum capacity and minimum water temperature limitations.

8. Check the data of Table XII for Freon 12 as was done for ammonia in Art. 309.

9. A single-cylinder Freon 12 compressor functions to give a refrigeration capacity of 20 tons under the following conditions: suction pressure, 26.51 psia; suction temperature, 14 F; condenser pressure, 107.9 psia; condenser temperature, 86 F; and Freon 12 liquid temperature, 77 F. Find (*a*) the weight of refrigerant supplied to the evaporator, pounds per minute; (*b*) the work of isentropic compression based on enthalpy change in the compressor, in Btu per minute, when $k = 1.14$; (*c*) the weight of condenser cooling water, in pounds per minute, when its temperature rise is 15 F; (*d*) the isentropic horsepower, hp_i, produced in the compressor cylinder; (*e*) the brake horsepower of the compressor if n_c is 0.80; and (*f*) the actual coefficient of performance.

10. Compute the isentropic horsepower per pound of Freon 22 for its compression under conditions stated in Table XII.

11. A belt-driven compressor operates to furnish a refrigerant to reduce the temperature of 200 gpm of water from 55 to 35 F. The belt efficiency is 98 per cent, the motor efficiency 88 per cent, and the input to the compressor 0.90 bhp per ton of refrigeration. (*a*) Find the tons of refrigeration capacity produced. (*b*) Find the actual coefficient of performance. (*c*) Find the weight, per hour, of condenser cooling water warmed from 70 to 90 F if 35 per cent of the heat equivalent of the power supplied to the compressor represents friction and cylinder cooling losses. (*d*) Find power input to motor, kilowatts per hour.

12. Based on discussions of clearance and volumetric efficiency in Chapter 14, dealing with the compression of air and gases, are compressors designed for use with refrigerants affected by these items when the compressors are in operation?

Appendix

TABLE XIII

Properties of Water*

Temperature, F	Boiling or Saturation Pressure, lb per sq in. abs	Volume, cu ft per lb	Weight, lb per cu ft	Enthalpy above 32 F, Btu per lb	Temperature, F	Boiling or Saturation Pressure, lb per sq in. abs	Volume, cu ft per lb	Weight, lb per cu ft	Enthalpy above 32 F, Btu per lb
32	0 08854	0 01602	62 42	0 0	135	2 5370	0 01627	61 46	102 90
35	0 09995	0 01602	62 42	3 02	140	2 8886	0 01629	61.39	107 89
40	0 12170	0 01602	62 42	8 05	145	3 281	0 01632	61.27	112 89
45	0 14752	0 01602	62 42	13 06	150	3 718	0 01634	61.20	117.89
50	0 17811	0 01603	62 38	18 07	155	4 203	0 01637	61.09	122.89
55	0 2141	0 01603	62 38	23 07	160	4 741	0 01639	61 01	127 89
60	0 2563	0 01604	62 34	28 06	165	5 335	0 01642	60 90	132.89
65	0 3056	0 01605	62 30	33 05	170	5 992	0 01645	60.79	137.90
70	0 3631	0 01606	62 27	38 04	175	6 715	0 01648	60 68	142 91
75	0 4298	0 01607	62 23	43 03	180	7 510	0 01651	60.57	147.92
80	0 5069	0 01608	62 19	48 02	185	8 383	0 01654	60 46	152.93
85	0 5959	0 01609	62 15	53 00	190	9 339	0 01657	60 35	157.95
90	0 6982	0 01610	62 11	57 99	195	10 385	0 01660	60.24	162.97
95	0 8153	0 01612	62 03	62 98	200	11 526	0 01663	60.13	167.99
100	0 9492	0 01613	61 99	67 97	205	12 771	0 01666	60.02	173.02
105	1 1016	0 01615	61 92	72 95	210	14 123	0 01670	59 88	178.05
110	1 2748	0 01617	61 84	77 94	215	15 595	0 01673	59.77	183.09
115	1 4709	0 01618	61 80	82 93	220	17 186	0 01677	59.63	188.13
120	1 6924	0 01620	61 73	87 92	225	18 916	0 01681	59 48	193.17
125	1 9420	0 01622	61 65	92 91	230	20 780	0 01684	59.38	198 23
130	2 2225	0 01625	61 54	97 90	240	24 969	0 01692	59.10	208 34

* From "Thermodynamic Properties of Steam," by J H Keenan and F. G. Keyes. Copyrighted 1936. John Wiley & Sons, Inc , New York, Publisher.

TABLE XIV

Dry Saturated Steam Temperature Table*

Temp, F	Abs Press, Lb Sq In	Specific Volume			Enthalpy			Entropy			Temp, F
		Sat Liquid	Evap	Sat Vapor	Sat Liquid	Evap	Sat Vapor	Sat Liquid	Evap	Sat Vapor	
t	p	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	t
32	0 08854	0 01602	3306	3306	0 00	1075 8	1075 8	0 0000	2 1877	2 1877	**32**
35	0 09995	0 01602	2947	2947	3 02	1074 1	1077 1	0 0061	2 1709	2 1770	**35**
40	0 12170	0 01602	2444	2444	8 05	1071 3	1079 3	0 0162	2 1435	2 1597	**40**
45	0 14752	0 01602	2036 4	2036 4	13 06	1068 4	1081 5	0 0262	2 1167	2 1429	**45**
50	0 17811	0 01603	1703 2	1703 2	18 07	1065 6	1083 7	0 0361	2 0903	2 1264	**50**
60	0 2563	0 01604	1206 6	1206 7	28 06	1059 9	1088 0	0 0555	2 0393	2 0948	**60**
70	0 3631	0 01606	867 8	867 9	38 04	1054 3	1092 3	0 0745	1 9902	2 0647	**70**
80	0 5069	0 01608	633 1	633 1	48 02	1048 6	1096 6	0 0932	1 9428	2 0360	**80**
90	0 6982	0 01610	468 0	468 0	57 99	1042 9	1100 9	0 1115	1 8972	2 0087	**90**
100	0 9492	0 01613	350 3	350 4	67 97	1037 2	1105 2	0 1295	1 8531	1 9826	**100**
110	1 2748	0 01617	265 3	265 4	77 94	1031 6	1109 5	0 1471	1 8106	1 9577	**110**
120	1 6924	0 01620	203 25	203 27	87 92	1025 8	1113 7	0 1645	1 7694	1 9339	**120**
130	2 2225	0 01625	157 32	157 34	97 90	1020 0	1117 9	0 1816	1 7296	1 9112	**130**
140	2 8886	0 01629	122 99	123 01	107 89	1014 1	1122 0	0 1984	1 6910	1 8894	**140**
150	3 718	0 01634	97 06	97 07	117 89	1008 2	1126 1	0 2149	1 6537	1 8685	**150**
160	4 741	0 01639	77 27	77 29	127 89	1002 3	1130 2	0 2311	1 6174	1 8485	**160**
170	5 992	0 01645	62 04	62 06	137 90	996 3	1134 2	0 2472	1 5822	1 8293	**170**
180	7 510	0 01651	50 21	50 23	147 92	990 2	1138 1	0 2630	1 5480	1 8109	**180**
190	9 339	0 01657	40 94	40 96	157 95	984 1	1142 0	0 2785	1 5147	1 7932	**190**
200	11 526	0 01663	33 62	33 64	167 99	977 9	1145 9	0 2938	1 4824	1 7762	**200**
210	14 123	0 01670	27 80	27 82	178 05	971 6	1149 7	0 3090	1 4508	1 7598	**210**
212	14 696	0 01672	26 78	26 80	180 07	970 3	1150 4	0 3120	1 4446	1 7566	**212**
220	17 186	0 01677	23 13	23 15	188 13	965 2	1153 4	0 3239	1 4201	1 7440	**220**
230	20 780	0 01684	19 365	19 382	198 23	958 8	1157 0	0 3387	1 3901	1 7288	**230**
240	24 969	0 01692	16 306	16 323	208 34	952 2	1160 5	0 3531	1 3609	1 7140	**240**
250	29 825	0 01700	13 804	13 821	218 48	945 5	1164 0	0 3675	1 3323	1 6998	**250**
260	35 429	0 01709	11 746	11 763	228 64	938 7	1167 3	0 3817	1 3043	1 6860	**260**
270	41 858	0 01717	10 044	10 061	238 84	931 8	1170 6	0 3958	1 2769	1 6727	**270**
280	49 203	0 01726	8 628	8 645	249 06	924 7	1173 8	0 4096	1 2501	1 6597	**280**
290	57 556	0 01735	7 444	7 461	259 31	917 5	1176 8	0 4234	1 2238	1 6472	**290**

300	67 013	0 01745	6 449	6 466	269 59	910 1	1179 7	0 4369	1 1980	1 6350	**300**
310	77 68	0 01755	5 609	5 626	279 92	902 6	1182 5	0 4504	1 1727	1 6231	**310**
320	89 66	0 01765	4 896	4 914	290 28	894 9	1185 2	0 4637	1 1478	1 6115	**320**
330	103 06	0 01776	4 289	4 307	300 68	887 0	1187 7	0 4769	1 1233	1 6002	**330**
340	118 01	0 01787	3 770	3 788	311 13	879 0	1190 1	0 4900	1 0992	1 5891	**340**
350	134 63	0 01799	3 324	3 342	321 63	870 7	1192 3	0 5029	1 0754	1 5783	**350**
360	153 04	0 01811	2 939	2 957	332 18	862 2	1194 4	0 5158	1 0519	1 5677	**360**
370	173 37	0 01823	2 606	2 625	342 79	853 5	1196 3	0 5286	1 0287	1 5573	**370**
380	195 77	0 01836	2 317	2 335	353 45	844 6	1198 1	0 5413	1 0059	1 5471	**380**
390	220 37	0 01850	2 0651	2 0836	364 17	835 4	1199 6	0 5539	0 9832	1 5371	**390**
400	247 31	0 01864	1 8447	1 8633	374 97	826 0	1201 0	0 5664	0 9608	1 5272	**400**
410	276 75	0 01878	1 6512	1 6700	385 83	816 3	1202 1	0 5788	0 9386	1 5174	**410**
420	308 83	0 01894	1 4811	1 5000	396 77	806 3	1203 1	0 5912	0 9166	1 5078	**420**
430	343 72	0 01910	1 3308	1 3499	407 79	796 0	1203 8	0 6035	0 8947	1 4982	**430**
440	381 59	0 01926	1 1979	1 2171	418 90	785 4	1204 3	0 6158	0 8730	1 4887	**440**
450	422 6	0 0194	1 0799	1 0993	430 1	774 5	1204 6	0 6280	0 8513	1 4793	**450**
460	466 9	0 0196	0 9748	0 9944	441 4	763 2	1204 6	0 6402	0 8298	1 4700	**460**
470	514 7	0 0198	0 8811	0 9009	452 8	751 5	1204 3	0 6523	0 8083	1 4606	**470**
480	566 1	0 0200	0 7972	0 8172	464 4	739 4	1203 7	0 6645	0 7868	1 4513	**480**
490	621 4	0 0202	0 7221	0 7423	476 0	726 8	1202 8	0 6766	0 7653	1 4419	**490**
500	680 8	0 0204	0 6545	0 6749	487 8	713 9	1201 7	0 6887	0 7438	1 4325	**500**
520	812 4	0 0209	0 5385	0 5594	511 9	686 4	1198 2	0 7130	0 7006	1 4136	**520**
540	962 5	0 0215	0 4434	0 4649	536 6	656 6	1193 2	0 7374	0 6568	1 3942	**540**
560	1133 1	0 0221	0 3647	0 3868	562 2	624 2	1186 4	0 7621	0 6121	1 3742	**560**
580	1325 8	0 0228	0 2989	0 3217	588 9	588 4	1177 3	0 7872	0 5659	1 3532	**580**
600	1542 9	0 0236	0 2432	0 2668	617 0	548 5	1165 5	0 8131	0 5176	1 3307	**600**
620	1786 6	0 0247	0 1955	0 2201	646 7	503 6	1150 3	0 8398	0 4664	1 3062	**620**
640	2059 7	0 0260	0 1538	0 1798	678 6	452 0	1130 5	0 8679	0 4110	1 2789	**640**
660	2365 4	0 0278	0 1165	0 1442	714 2	390 2	1104 4	0 8987	0 3485	1 2472	**660**
680	2708 1	0 0305	0 0810	0 1115	757 3	309 9	1067 2	0 9351	0 2719	1 2071	**680**
700	3093 7	0 0369	0 0392	0 0761	823 3	172 1	995 4	0 9905	0 1484	1 1389	**700**
705 4	3206 2	0 0503	0	0 0503	902 7	0	902 7	1 0580	0	1 0580	**705 4**

* Abridged from "Thermodynamic Properties of Steam" by Joseph H. Keenan and Frederick G Keyes Copyright, 1936, by Joseph H Keenan and Frederick G Keyes Published by John Wiley & Sons, Inc , New York.

TABLE XV

Dry Saturated Steam. Pressure Table*

Abs Press., Lb Sq In.	Temp., F	Specific Volume: Sat Liquid	Specific Volume: Sat Vapor	Enthalpy: Sat Liquid	Enthalpy: Evap	Enthalpy: Sat Vapor	Entropy: Sat Liquid	Entropy: Evap	Entropy: Sat Vapor	Internal Energy: Sat Liquid	Internal Energy: Sat Vapor	Abs Press., Lb Sq In.
p	*t*	v_f	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	u_f	u_g	*p*
0 491	79 03	0 01608	652 3	47 05	1049 2	1096 3	0 0914	1 9473	2 0387	47 05	1037 0	**0 491**
0 736	91 72	0 01611	444 9	59 71	1042 0	1101 7	0 1147	1 8894	2 0041	59 71	1041 1	**0 736**
0 982	101 14	0 01614	339 2	69 10	1036 6	1105 7	0 1316	1 8481	1 9797	69 10	1044 0	**0 982**
1 227	108 71	0 01616	274 9	76 65	1032 3	1108 9	0 1449	1 8160	1 9609	76 65	1046 4	**1 227**
1 473	115 06	0 01618	231 6	82 99	1028 6	1111 6	0 1560	1 7896	1 9456	82 99	1048 5	**1 473**
1 964	125 43	0 01622	176 7	93 34	1022 7	1116 0	0 1738	1 7476	1 9214	93 33	1051 8	**1 964**
2 455	133 76	0 01626	143 25	101 66	1017 7	1119 4	0 1879	1 7150	1 9028	101 65	1054 3	**2 455**
5	162 24	0 01640	73 52	130 13	1001 0	1131 1	0 2347	1 6094	1 8441	130 12	1063 1	**5**
10	193 21	0 01659	38 42	161 17	982 1	1143 3	0 2835	1 5041	1 7876	161 14	1072 2	**10**
14 696	212 0	0 01672	26 80	180 07	970 3	1150 4	0 3120	1 4446	1 7566	180 02	1077 5	**14 696**
15	213 03	0 01672	26 29	181 11	969 7	1150 8	0 3135	1 4415	1 7549	181 06	1077 8	**15**
16	216 32	0 01674	24 75	184 42	967 6	1152 0	0 3184	1 4313	1 7497	184 37	1078 7	**16**
18	222 41	0 01679	22 17	190 56	963 6	1154 2	0 3275	1 4128	1 7403	190 50	1080 4	**18**
20	227 96	0 01683	20 089	196 16	960 1	1156 3	0 3356	1 3962	1 7319	196 10	1081 9	**20**
25	240 07	0 01692	16 303	208 42	952 1	1160 6	0 3533	1 3606	1 7139	208 34	1085 1	**25**
30	250 33	0 01701	13 746	218 82	945 3	1164 1	0 3680	1 3313	1 6993	218 73	1087 8	**30**
35	259 28	0 01708	11 898	227 91	939 2	1167 1	0 3807	1 3063	1 6870	227 80	1090 1	**35**
40	267 25	0 01715	10 498	236 03	933 7	1169 7	0 3919	1 2844	1 6763	235 90	1092 0	**40**
45	274 44	0 01721	9 401	243 36	928 6	1172 0	0 4019	1 2650	1 6669	243 22	1093 7	**45**
50	281 01	0 01727	8 515	250 09	924 0	1174 1	0 4110	1 2474	1 6585	249 93	1095 3	**50**
55	287 07	0 01732	7 787	256 30	919 6	1175 9	0 4193	1 2316	1 6509	256 12	1096 7	**55**
60	292 71	0 01738	7 175	262 09	915 5	1177 6	0 4270	1 2168	1 6438	261 90	1097 9	**60**
65	297 97	0 01743	6 655	267 50	911 6	1179 1	0 4342	1 2032	1 6374	267 29	1099 1	**65**
70	302 92	0 01748	6 206	272 61	907 9	1180 6	0 4409	1 1906	1 6315	272 38	1100 2	**70**
75	307 60	0 01753	5 816	277 43	904 5	1181 9	0 4472	1 1787	1 6259	277 19	1101 2	**75**
80	312 03	0 01757	5 472	282 02	901 1	1183 1	0 4531	1 1676	1 6207	281 76	1102 1	**80**
85	316 25	0 01761	5 168	286 39	897 8	1184 2	0 4587	1 1571	1 6158	286 11	1102 9	**85**
90	320 27	0 01766	4 896	290 56	894 7	1185 3	0 4641	1 1471	1 6112	290 27	1103 7	**90**
100	327 81	0 01774	4 432	298 40	888 8	1187 2	0 4740	1 1286	1 6026	298 08	1105 2	**100**
110	334 77	0 01782	4 049	305 66	883 2	1188 9	0 4832	1 1117	1 5948	305 30	1106 5	**110**

120	341 25	0 01789	3 728	312 44	877 9	1190 4	0 4916	1 0962	1 5878	312 05	1107 6	**120**
130	347 32	0 01796	3 455	318 81	872 9	1191 7	0 4995	1 0817	1 5812	318 38	1108 6	**130**
140	353 02	0 01802	3 220	324 82	868 2	1193 0	0 5069	1 0682	1 5751	324 35	1109 6	**140**
150	358 42	0 01809	3 015	330 51	863 6	1194 1	0 5138	1 0556	1 5694	330 01	1110 5	**150**
160	363 53	0 01815	2 834	335 93	859 2	1195 1	0 5204	1 0436	1 5640	335 39	1111 2	**160**
170	368 41	0 01822	2 675	341 09	854 9	1196 0	0 5266	1 0324	1 5590	340 52	1111 9	**170**
180	373 06	0 01827	2 532	346 03	850 8	1196 9	0 5325	1 0217	1 5542	345 42	1112 5	**180**
190	377 51	0 01833	2 404	350 79	846 8	1197 6	0 5381	1 0116	1 5497	350 15	1113 1	**190**
200	381 79	0 01839	2 288	355 36	843 0	1198 4	0 5435	1 0018	1 5453	354 68	1113 7	**200**
250	400 95	0 01865	1 8438	376 00	825 1	1201 1	0 5675	0 9588	1 5263	375 14	1115 8	**250**
300	417 33	0 01890	1 5433	393 84	809 0	1202 8	0 5879	0 9225	1 5104	392 79	1117 1	**300**
350	431 72	0 01913	1 3260	409 69	794 2	1203 9	0 6056	0 8910	1 4966	408 45	1118 0	**350**
400	444 59	0 0193	1 1613	424 0	780 5	1204 5	0 6214	0 8630	1 4844	422 6	1118 5	**400**
450	456 28	0 0195	1 0320	437 2	767 4	1204 6	0 6356	0 8378	1 4734	435 5	1118 7	**450**
500	467 01	0 0197	0 9278	449 4	755 0	1204 4	0 6487	0 8147	1 4634	447 6	1118 6	**500**
550	476 94	0 0199	0 8424	460 8	743 1	1203 9	0 6608	0 7934	1 4542	458 8	1118 2	**550**
600	486 21	0 0201	0 7698	471 6	731 6	1203 2	0 6720	0 7734	1 4454	469 4	1117 7	**600**
650	494 90	0 0203	0 7083	481 8	720 5	1202 3	0 6826	0 7548	1 4374	479 4	1117 1	**650**
700	503 10	0 0205	0 6554	491 5	709 7	1201 2	0 6925	0 7371	1 4296	488 8	1116 3	**700**
750	510 86	0 0207	0 6092	500 8	699 2	1200 0	0 7019	0 7204	1 4223	598 0	1115 4	**750**
800	518 23	0 0209	0 5687	509 7	688 9	1198 6	0 7108	0 7045	1 4153	506 6	1114 4	**800**
850	525 26	0 0210	0 5327	518 3	678 8	1197 1	0 7194	0 6891	1 4085	515 0	1113 3	**850**
900	531 98	0 0212	0 5006	526 6	668 8	1195 4	0 7275	0 6744	1 4020	523 1	1112 1	**900**
950	538 43	0 0214	0 4717	534 6	659 1	1193 7	0 7355	0 6602	1 3957	530 9	1110 8	**950**
1000	544 61	0 0216	0 4456	542 4	649 4	1191 8	0 7430	0 6467	1 3897	538 4	1109 4	**1000**
1100	556 31	0 0220	0 4001	557 4	630 4	1187 8	0 7575	0 6205	1 3780	552 9	1106 4	**1100**
1200	567 22	0 0223	0 3619	571 7	611 7	1183 4	0 7711	0 5956	1 3667	566 7	1103 0	**1200**
1300	577 46	0 0227	0 3293	585 4	593 2	1178 6	0 7840	0 5719	1 3559	580 0	1099 4	**1300**
1400	587 10	0 0231	0 3012	598 7	574 7	1173 4	0 7963	0 5491	1 3454	592 7	1095 4	**1400**
1500	596 23	0 0235	0 2765	611 6	556 3	1167 9	0 8082	0 5269	1 3351	605 1	1091 2	**1500**
2000	635 82	0 0257	0 1878	671 7	463 4	1135 1	0 8619	0 4230	1 2849	662 2	1065 6	**2000**
2500	668 13	0 0287	0 1307	730 6	360 5	1091 1	0 9126	0 3197	1 2322	717 3	1030 6	**2500**
3000	695 36	0 0346	0 0858	802 5	217 8	1020 3	0 9731	0 1885	1 1615	783 4	972 7	**3000**
3206 2	705 40	0 0503	0 0503	902 7	0	902 7	1 0580	0	1 0580	872 9	872 9	**3206.2**

TABLE XVI

Properties of Superheated Steam*

Abs Press, Lb Sq In (Sat Temp)		Temperature—Degrees Fahrenheit 200	220	300	350	400	450	500	550	600	700	800	900	1000
1	*v*	392 6	404 5	452 3	482 2	512 0	541 8	571 6	601 4	631 2	690 8	750 4	809 9	869 5
	h	1150 4	1159 5	1195 8	1218 7	1241 7	1264 9	1288 3	1312 0	1335 7	1383 8	1432 8	1482 7	1533 5
(101 74)	*s*	2 0512	2 0647	2 1153	2 1444	2 1720	2 1983	2 2233	2 2468	2 2702	2 3137	2 3542	2 3923	2 4283
5	*v*	78 16	80 59	90 25	96 26	102 26	108 24	114 22	120 19	126 16	138 10	150 03	161 95	173 87
	h	1148 8	1158 1	1195 0	1218 1	1241 2	1264 5	1288 0	1311 7	1335 4	1383 6	1432 7	1482 6	1533 4
(162 24)	*s*	1 8718	1 8857	1 9370	1 9664	1 9942	2 0205	2 0456	2 0692	2 0927	2 1361	2 1767	2 2148	2 2509
10	*v*	38 85	40 09	45 00	48 03	51 04	54 05	57 05	60 04	63 03	69 01	74 98	80 95	86 92
	h	1146 6	1156 2	1193 9	1217 2	1240 6	1264 0	1287 5	1311 3	1335 1	1383 4	1432 5	1482 4	1533 2
(193 21)	*s*	1 7927	1 8071	1 8595	1 8892	1 9172	1 9436	1 9689	1 9924	2 0160	2 0596	2 1002	2 1383	2 1744
14 696	*v*		27 15	30 53	32 62	34 68	36 73	38 78	40 82	42 86	46 94	51 00	55 07	59 13
	h		1154 4	1192 8	1216 4	1239 9	1263 5	1287 1	1310 9	1334 8	1383 2	1432 3	1482 3	1533 1
(212 00)	*s*		1 7624	1 8160	1 8460	1 8743	1 9008	1 9261	1 9498	1 9734	2 0170	2 0576	2 0958	2 1319
20	*v*			22 36	23 91	25 43	26 95	28 46	29 97	31 47	34 47	37 46	40 45	43 44
	h			1191 6	1215 6	1239 2	1262 9	1286 6	1310 5	1334 4	1382 9	1432 1	1482 1	1533 0
(227 96)	*s*			1 7808	1 8112	1 8396	1 8664	1 8918	1 9160	1 9392	1 9829	2 0235	2 0618	2 0978
40	*v*			11 040	11 843	12 628	13 401	14 168	14 93	15 688	17 198	18 702	20 20	21 70
	h			1186 8	1211 9	1236 5	1260 7	1284 8	1308 9	1333 1	1381 9	1431 3	1481 4	1532 4
(267 25)	*s*			1 6994	1 7314	1 7608	1 7881	1 8140	1 8384	1 8619	1 9058	1 9467	1 9850	2 0214
60	*v*			7 259	15 635	8 357	8 884	9 403	9 916	10 427	11 441	12 449	13 452	14 454
	h			1181 6	1208 2	1233 6	1258 5	1283 0	1307 4	1331 8	1380 9	1430 5	1480 8	1531 9
(292 71)	*s*			1 6492	1 6830	1 7135	1 7416	1 7678	1 7926	1 8162	1 8605	1 9015	1 9400	1 9762
80	*v*				5 803	6 22	6 624	7 020	7 410	7 797	8 562	9 322	10 077	10 830
	h				1204 3	1230 7	1256 1	1281 1	1305 8	1330 5	1379 9	1429 7	1480 1	1531 3
(312 03)	*s*				1 6475	1 6791	1 7078	1 7346	1 7598	1 7836	1 8281	1 8694	1 9079	1 9442
100	*v*				4 592	4 937	5 268	5 589	5 905	6 218	6 835	7 446	8 052	8 656
	h				1200 1	1227 6	1253 7	1279 1	1304 2	1329 1	1378 9	1428 9	1479 5	1530 8
(327 81)	*s*				1 6188	1 6518	1 6813	1 7085	1 7339	1 7581	1 8029	1 8443	1 8829	1 9193
120	*v*				3 783	4 081	4 363	4 636	4 902	5 165	5 683	6 195	6 702	7 207
	h				1195 7	1224 4	1251 3	1277 2	1302 5	1327 7	1377 8	1428 1	1478 8	1530 2
(341 25)	*s*				1 5944	1 6287	1 6591	1 6869	1 7127	1 7370	1 7822	1 8237	1 8625	1 8990

	v	3 468	3 715	3 954	4 186	4 413	4 861	5 301	5 738	6 172
140	*h*	1221 1	1248 7	1275 2	1300 9	1326 4	1376 8	1427 3	1478 2	1529 7
(353 02)	*s*	1 6087	1 6515	1 6683	1 6945	1 7190	1 7645	1 8063	1 8451	1 8817
	v	3 008	3 230	3 443	3 648	3 849	4 244	4 631	5 015	5 396
160	*h*	1217 6	1246 1	1273 1	1299 3	1325 0	1375 7	1426 4	1477 5	1529 1
(363 53)	*s*	1 5908	1 6230	1 6519	1 6785	1 7033	1 7491	1 7911	1 8301	1 8667
	v	2 649	2 852	3 044	3 229	3 411	3 764	4 110	4 452	4 792
180	*h*	1214 0	1243 5	1271 0	1297 6	1323 5	1374 7	1425 6	1476 8	1528 6
(373 06)	*s*	1 5745	1 6077	1 6373	1 6642	1 6894	1 7355	1 7776	1 8167	1 8534
	v	2 361	2 549	2 726	2 895	3 060	3 380	3 693	4 002	4 309
200	*h*	1210 3	1240 7	1268 9	1295 8	1322 1	1373 6	1424 8	1476 2	1528.0
(381 79)	*s*	1 5594	1 5937	1 6240	1 6513	1 6767	1 7232	1 7655	1 8048	1 8415
	v	2 125	2 301	2 465	2 621	2 772	3 066	3 352	3 634	3 913
220	*h*	1206 5	1237 9	1266 7	1294 1	1320 7	1372 6	1424 0	1475 5	1527 5
(389 86)	*s*	1 5453	1 5808	1 6117	1 6395	1 6652	1 7120	1 7545	1 7939	1 8308
	v	1 9276	2 1120	2 247	2 393	2 533	2 804	3 068	3 327	3 584
240	*h*	1202 5	1234 9	1264 5	1292 4	1319 2	1371 5	1423 2	1474 8	1526 9
(397 37)	*s*	1 5319	1 5686	1 6003	1 6286	1 6546	1 7017	1 7444	1 7839	1 8209
	v		1 9183	2 063	2 199	2 330	2 582	2 827	3 067	3 305
260	*h*		1232 0	1262 3	1290 5	1317 7	1370 4	1422 3	1474 2	1526 3
(404 42)	*s*		1 5573	1 5897	1 6184	1 6447	1 6922	1 7352	1 7748	1 8118
	v		1 7674	1 9047	2 033	2 156	2 392	2 621	2 845	3 066
280	*h*		1228 9	1260 0	1288 7	1316 2	1369 4	1421 5	1473 5	1525 8
(411 05)	*s*		1 5464	1 5796	1 6087	1 6354	1 6834	1 7265	1 7662	1 8033
	v		1 6364	1 7675	1 8891	2 005	2 227	2 442	2 652	2 859
300	*h*		1225 8	1257 6	1286 8	1314 7	1368 3	1420 6	1472 8	1525 2
(417 33)	*s*		1 5360	1 5701	1 5998	1 6268	1 6751	1 7184	1 7582	1 7954
	v		1 3734	1 4923	1 6010	1 7036	1 8980	2 084	2 266	2 445
350	*h*		1217 7	1251 5	1282 1	1310 9	1365 5	1418 5	1471 1	1523 8
(431 72)	*s*		1 5119	1 5481	1 5792	1 6070	1 6563	1 7002	1 7403	1 7777
	v		1 1744	1 2851	1 3843	1 4770	1 6508	1 8161	1 9767	2 134
400	*h*		1208 8	1245 1	1277 2	1306 9	1362 7	1416 4	1469 4	1522 4
(444 59)	*s*		1 4892	1 5281	1 5607	1 5894	1 6398	1 6842	1 7247	1 7623

TABLE XVI—*Continued*

Properties of Superheated Steam*

Abs Press. Lb Sq In. (Sat. Temp.)		Temperature—Degrees Fahrenheit 500	550	600	620	640	660	680	700	800	900	1000	1200	1400	1600
	v	1.1231	1.2155	1.3005	1.3332	1.3652	1.3967	1.4278	1.4584	1.6074	1.7516	1.8928	2.170	2.443	2.714
450	h	1238.4	1272.0	1302.8	1314.6	1326.2	1337.5	1348.8	1359.9	1414.3	1467.7	1521.0	1628.6	1738.7	1851.9
(456.28)	s	1.5095	1.5437	1.5735	1.5845	1.5951	1.6054	1.6153	1.6250	1.6699	1.7108	1.7486	1.8177	1.8803	1.9381
	v	0.9927	1.0800	1.1591	1.1893	1.2188	1.2478	1 2763	1.3044	1.4405	1.5715	1.6996	1.9504	2.197	2.442
500	h	1231.3	1266.8	1298.6	1310.7	1322.6	1334.2	1345.7	1357.0	1412.1	1466.0	1519.6	1627.6	1737.9	1851.3
(467.01)	s	1.4919	1.5280	1.5588	1.5701	1.5810	1.5915	1.6016	1.6115	1.6571	1.6982	1.7363	1.8056	1.8683	1.9262
	v	0.8852	0.9686	1.0431	1.0714	1.0989	1.1259	1.1523	1.1783	1.3038	1.4241	1.5414	1.7706	1.9957	2.219
550	h	1223.7	1261.2	1294.3	1306.8	1318.9	1330.8	1342.5	1354.0	1409.9	1464.3	1518.2	1626.6	1737.1	1850.6
(476.94)	s	1.4751	1.5131	1.5451	1.5568	1.5680	1.5787	1.5890	1.5991	1.6452	1.6868	1.7250	1.7946	1.8575	1.9155
	v	0.7947	0.8753	0.9463	0.9729	0.9988	1.0241	1.0489	1.0732	1.1899	1.3013	1.4096	1.6208	1.8279	2.033
600	h	1215.7	1255.5	1289.9	1302.7	1315.2	1327.4	1339.3	1351.1	1407.7	1462.5	1516.7	1625.5	1736.3	1850.0
(486.21)	s	1.4586	1.4990	1.5323	1.5443	1.5558	1.5667	1.5773	1.5875	1.6343	1.6762	1.7147	1.7846	1.8476	1.9056
	v	......	0.7277	0.7934	0.8177	0.8411	0.8639	0 8860	0.9077	1.0108	1.1082	1.2024	1.3853	1.5641	1.7405
700	h	......	1243.2	1280.6	1294.3	1307.5	1320 3	1332.8	1345.0	1403.2	1459.0	1513.9	1623.5	1734.8	1848.8
(503.10)	s	......	1.4722	1.5084	1.5212	1.5333	1.5449	1.5559	1.5665	1.6147	1.6573	1.6963	1.7666	1.8299	1.8881
	v	......	0.6154	0.6779	0.7006	0.7223	0.7433	0 7635	0.7833	0.8763	0.9633	1.0470	1.2088	1.3662	1.5214
800	h	......	1229.8	1270.7	1285.4	1299.4	1312.9	1325.9	1338.6	1398.6	1455.4	1511.0	1621.4	1733.2	1847.5
(518.23)	s	......	1.4467	1.4863	1.5000	1.5129	1.5250	1.5366	1.5476	1.5972	1.6407	1.6801	1.7510	1.8146	1.8729
	v	......	0.5264	0.5873	0 6089	0.6294	0 6491	0.6680	0.6863	0.7716	0.8506	0.9262	1.0714	1.2124	1.3509
900	h	......	1215.0	1260.1	1275.9	1290.9	1305.1	1318.8	1332.1	1393.9	1451.8	1508.1	1619.3	1731.6	1846.3
(531.98)	s	......	1.4216	1.4653	1.4800	1.4938	1.5066	1.5187	1.5303	1.5814	1.6257	1.6656	1.7371	1.8009	1.8595
	v	......	0.4533	0.5140	0.5350	0.5546	0.5733	0.5912	0.6084	0.6878	0.7604	0.8294	0.9615	1.0893	1.2146
1000	h	......	1198.3	1248.8	1265.9	1281.9	1297.0	1311.4	1325.3	1389.2	1448.2	1505.1	1617.3	1730.0	1845.0
(544.61)	s	......	1.3961	1.4450	1.4610	1.4757	1.4893	1.5021	1.5141	1.5670	1.6121	1.6525	1.7245	1.7886	1.8474
	v	......	......	0.4532	0 4738	0.4929	0.5110	0 5281	0 5445	0.6191	0.6866	0.7503	0.8716	0.9885	1.1031
1100	h	......	......	1236.7	1255 3	1272.4	1288.5	1303 7	1318.3	1384.3	1444.5	1502.2	1615.2	1728.4	1843.8
(556.31)	s	......	......	1.4251	1.4425	1.4583	1.4728	1.4862	1.4989	1.5535	1.5995	1.6405	1.7130	1.7775	1.8363
	v	......	......	0 4016	0 4222	0.4410	0.4586	0.4752	0.4909	0.5617	0.6250	0.6843	0.7967	0.9046	1.0101
1200	h	......	......	1223.5	1243.9	1262.4	1279.6	1295 7	1311.0	1379.3	1440.7	1499.2	1613.1	1726.9	1842.5
(567.22)	s	......	......	1.4052	1.4243	1.4413	1.4568	1.4710	1.4843	1.5409	1.5879	1.6293	1.7025	1.7672	1.8263

	v	......	......	0.3174	0.3390	0.3580	0.3753	0.3912	0.4062	0.4714	0.5281	0.5805	0.6789	0.7727	0.8640
1400	h	......	......	1193.0	1218.4	1240.4	1260.3	1278.5	1295.5	1369.1	1433.1	1493.2	1608.9	1723.7	1840.0
(587.10)	s	......	......	1.3639	1.3877	1.4079	1.4258	1.4419	1.4567	1.5177	1.5666	1.6093	1.6836	1.7489	1.8083
	v	......	......	......	0.2733	0.2936	0.3112	0.3271	0.3417	0.4034	0.4553	0.5027	0.5906	0.6738	0.7545
1600	h	......	......	......	1187.8	1215.2	1238.7	1259.6	1278.7	1358.4	1425.3	1487.0	1604.6	1720.5	1837.5
(604.90)	s	......	......	......	1.3489	1.3741	1.3952	1.4137	1.4303	1.4964	1.5476	1.5914	1.6669	1.7328	1.7926
	v	......	......	......	......	0.2407	0.2597	0.2760	0.2907	0.3502	0.3986	0.4421	0.5218	0.5968	0.6693
1800	h	......	......	......	......	1185.1	1214.0	1238.5	1260.3	1347.2	1417.4	1480.8	1600.4	1717.3	1835.0
(621.03)	s	......	......	......	......	1.3377	1.3638	1.3855	1.4044	1.4765	1.5301	1.5752	1.6520	1.7185	1.7786
	v	......	......	......	......	0.1936	0.2161	0.2337	0.2489	0.3074	0.3532	0.3935	0.4668	0.5352	0.6011
2000	h	......	......	......	......	1145.6	1184.9	1214.8	1240.0	1335.5	1409.2	1474.5	1596.1	1714.1	1832.5
(635.82)	s	......	......	......	......	1.2945	1.3300	1.3564	1.3783	1.4576	1.5139	1.5603	1.6384	1.7055	1.7660
	v	......	......	......	......	......	......	0.1484	0.1686	0.2294	0.2710	0.3061	0.3678	0.4244	0.4784
2500	h	......	......	......	......	......	......	1132.3	1176.8	1303.6	1387.8	1458.4	1585.3	1706.1	1826.2
(668.13)	s	......	......	......	......	......	......	1.2687	1.3073	1.4127	1.4772	1.5273	1.6088	1.6775	1.7389
	v	......	......	......	......	......	......	......	0.0984	0.1760	0.2159	0.2476	0.3018	0.3505	0.3966
3000	h	......	......	......	......	......	......	......	1060.7	1267.2	1365.0	1441.8	1574.3	1698.0	1819.9
(695.36)	s	......	......	......	......	......	......	......	1.1966	1.3690	1.4439	1.4984	1.5837	1.6540	1.7163
	v	......	......	......	......	......	......	......	......	0.1583	0.1981	0.2288	0.2806	0.3267	0.3703
3206.2	h	......	......	......	......	......	......	......	......	1250.5	1355.2	1434.7	1569.8	1694.6	1817.2
(705.40)	s	......	......	......	......	......	......	......	......	1.3508	1.4309	1.4874	1.5742	1.6452	1.7080
	v	......	......	......	......	......	......	......	0.0306	0.1364	0.1762	0.2058	0.2546	0.2977	0.3381
3500	h	......	......	......	......	......	......	......	780.5	1224.9	1340.7	1424.5	1563.3	1689.8	1813.6
	s	......	......	......	......	......	......	......	0.9515	1.3241	1.4127	1.4723	1.5615	1.6336	1.6968
	v	......	......	......	......	......	......	......	0.0287	0.1052	0.1462	0.1743	0.2192	0.2581	0.2943
4000	h	......	......	......	......	......	......	......	763.8	1174.8	1314.4	1406.8	1552.1	1681.7	1807.2
	s	......	......	......	......	......	......	......	0.9347	1.2757	1.3827	1.4482	1.5417	1.6154	1.6795
	v	......	......	......	......	......	......	......	0.0276	0.0798	0.1226	0.1500	0.1917	0.2273	0.2602
4500	h	......	......	......	......	......	......	......	753.5	1113.9	1286.5	1388.4	1540.8	1673.5	1800.9
	s	......	......	......	......	......	......	......	0.9235	1.2204	1.3529	1.4253	1.5235	1.5990	1.6640
	v	......	......	......	......	......	......	......	0.0268	0.0593	0.1036	0.1303	0.1696	0.2027	0.2329
5000	h	......	......	......	......	......	......	......	746.4	1047.1	1256.5	1369.5	1529.5	1665.3	1794.5
	s	......	......	......	......	......	......	......	0.9152	1.1622	1.3231	1.4034	1.5066	1.5839	1.6499
	v	......	......	......	......	......	......	......	0.0262	0.0463	0.0880	0.1143	0.1516	0.1825	0.2106
5500	h	......	......	......	......	......	......	......	741.3	985.0	1224.1	1349.3	1518.2	1657.0	1788.1
	s	......	......	......	......	......	......	......	0.9090	1.1093	1.2930	1.3821	1.4908	1.5699	1.6369

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TABLE XVII

Compressed Liquid*

The following data are for water

Pressure psia (Sat Temp F)		Temperature Degrees Fahrenheit 32	100	200	300	400	500	600
	Saturated Liquid P	0 08854	0 9492	11 526	67 013	247 31	680 8	1542 9
	v_f	0 016022	0 016132	0 016634	0 017449	0 018639	0 020432	0 023629
	h_f	0	67 97	167 99	269 59	374 97	487 82	617 0
	s_f	0	0 12948	0 29382	0 43694	0 56638	0 68871	0 8131
200	$(v - v_f)\ 10^5$	−1 1	−1 1	−1 1	−1 1			
(381 79)	$(h - h_f)$	+0 61	+0 54	+0 41	+0 23			
	$(s - s_f)\ 10^3$	+0 03	−0 05	−0 21	−0 21			
400	$(v - v_f)\ 10^5$	−2 3	−2 1	−2 2	−2 8	−2 1		
(444 59)	$(h - h_f)$	+1 21	+1 09	+0 88	+0 61	+0 16		
	$(s - s_f)\ 10^3$	+0 04	−0 16	−0 47	−0 56	−0 40		
600	$(v - v_f)\ 10^5$	−3 5	−3 2	−3 4	−4 3	−4 4		
(486 21)	$(h - h_f)$	+1 80	+1 67	+1 31	+0 97	+0 39		
	$(s - s_f)\ 10^3$	+0 07	−0 27	−0 74	−0 94	−0 96		
800	$(v - v_f)\ 10^5$	−4 6	−4 0	−4 4	−5 6	−6 5	−1 7	
(518 23)	$(h - h_f)$	+2 39	+2 17	+1 78	+1 35	+0 61	−0 05	
	$(s - s_f)\ 10^3$	+0 10	−0 40	−0 97	−1 27	−1 48	−0 53	
1000	$(v - v_f)\ 10^5$	−5 7	−5 1	−5 4	−6 9	−8 7	−6 4	
(544 61)	$(h - h_f)$	+2 99	+2 70	+2 21	+1 75	+0 84	−0 14	
	$(s - s_f)\ 10^3$	+0 15	−0 53	−1 20	−1 64	−2 00	−1 41	
1500	$(v - v_f)\ 10^5$	−8 4	−7 5	−8 1	−10 4	−14 1	−17 3	
(596 23)	$(h - h_f)$	+4 48	+3 99	+3 36	+2 70	+1 44	−0 29	
	$(s - s_f)\ 10^3$	+0 20	−0 86	−1 79	−2 53	−3 32	−3 56	
2000	$(v - v_f)\ 10^5$	−11 0	−9 9	−10 8	−13 8	−19 5	−27 8	−32 6
(635 82)	$(h - h_f)$	+5 97	+5 31	+4 51	+3 64	+2 03	−0 38	−2 5
	$(s - s_f)\ 10^3$	+0 22	−1 18	−2 39	−3 42	−4 57	−5 58	−4 3
2500	$(v - v_f)\ 10^5$	−13 7	−12 3	−13 4	−17 2	−24 8	−37 7	−61 9
(668 13)	$(h - h_f)$	+7 49	+6 58	+5 63	+4 55	+2 66	−0 41	−4 9
	$(s - s_f)\ 10^3$	+0 25	−1 48	−2 97	−4 25	−5 79	−7 54	−8 5
3000	$(v - v_f)\ 10^5$	−16 3	−14 7	−16 0	−20 7	−30 0	−47 1	−87 9
(695 36)	$(h - h_f)$	+9 00	+7 88	+6 76	+5 49	+3 33	−0 41	−6 9
	$(s - s_f)\ 10^3$	+0 28	−1 79	−3 56	−5 12	−7 03	−9 42	−12 4

* Abridged from Thermodynamic Properties of Steam by Joseph H Keenan and Frederick G. Keyes Copyright 1936 by Joseph H Keenan and Frederick G Keyes Published by John Wiley & Sons Inc New York. Used by permission

TABLE XVIII

Properties of Dry Air and Saturated Air

Absolute Pressure, 29.921 Inches of Mercury

Temperature, degrees F	Weight of 1 Cubic Foot of Dry Air, pound	Volume of 1 Pound of Dry Air, cubic feet	Volume of 1 Pound of Dry Air Plus Vapor to Saturate it, cubic feet	Weight of Water Necessary to Saturate 1 Cubic Foot of Dry Air, pound	Weight of Water Necessary to Saturate 1 Pound of Dry Air, pounds
0	0 08633	11 58	11 59	0 000068	0 000785
10	0 08449	11 83	11 85	0 000111	0 001311
20	0 08273	12 09	12 13	0 000189	0 002290
30	0 08104	12 34	12 41	0 000280	0 003479
40	0 07942	12 59	12 70	0 000410	0 005201
50	0 07785	12 84	13 00	0 000586	0 007632
60	0 07636	13 10	13 33	0 000828	0 01104
70	0 07492	13 35	13 69	0 001152	0 01576
80	0 07353	13 59	14 09	0 001576	0 02223
90	0 07219	13 85	14 54	0 002135	0 03104
100	0 07090	14 10	15 08	0 002845	0 04296
110	0 06966	14 36	15 72	0 003760	0 0591
120	0 06845	14 60	16 51	0 00490	0 0810
130	0 06729	14 86	17 51	0 00633	0 1110
140	0 06617	15 11	18 81	0 00814	0 1532
150	0 06509	15 36	20 57	0 01031	0 2122
160	0 06403	15 61	23 08	0 01295	0 2987
170	0 06295	15 89	26 80	0 01614	0 4324
180	0 06203	16 12	33 00	0 01991	0 6577
190	0 06109	16 38	45 10	0 2435	1 0985
200	0 06015	16 62	77 10	0 2975	2 2953

TABLE XIX

Dimensions of Schedules 30 and 40, Welded Steel Pipe for Steam, Air, Gas, and Water*

Nominal Size, in.	Inside Diameter, in.	External Diameter, in.	Nominal Thickness, in.	Transverse Areas			Length of Pipe Per Square Foot of External Surface, ft	Length of Pipe Containing One Cubic Foot, ft	Nominal Weight per Foot Plain Ends, lb	Number of Threads per Inch of Screw
				Internal, sq in.	External, sq in	Metal, sq in				
⅛	0 269	0 405	**0.068**	0 057	0 129	0 072	9 431	2533 775	0 244	27
¼	0 364	0 540	**0.088**	0 104	0 229	0 125	7 073	1383 789	0 424	18
⅜	0 493	0 675	**0.091**	0 191	0 358	0 167	5 658	754 360	0 567	18
½	0 622	0 840	**0.109**	0 304	0 554	0 250	4 547	473 906	0 850	14
¾	0 824	1 050	**0.113**	0 533	0 866	0 333	3 637	270 034	1 130	14
1	1 049	1 315	**0.133**	0 864	1 358	0 494	2 904	166 618	1 678	11½
1¼	1 380	1 660	**0.140**	1 495	2 164	0 669	2 301	96 275	2 272	11½
1½	1 610	1 900	**0.145**	2 036	2 835	0 799	2 010	70 733	2 717	11½
2	2 067	2 375	**0.154**	3 355	4 430	1 075	1 608	42 913	3 652	11½
2½	2 469	2 875	**0.203**	4 788	6 492	1 704	1 328	30 077	5 793	8
3	3 068	3 500	**0.216**	7 393	9 621	2 228	1 091	19.479	7 575	8
3½	3 548	4 000	**0.226**	9 886	12 566	2 680	0 954	14 565	9 109	8
4	4 026	4 500	**0.237**	12 730	15 904	3 174	0 848	11 312	10 790	8
5	5 047	5 563	**0.258**	20 006	24 306	4 300	0 686	7 198	14 617	8
6	6 065	6 625	**0.280**	28 891	34 472	5 581	0 576	4 984	18 974	8
8†	8 071	8 625	0.277	51 161	58 426	7 265	0 443	2 815	24 696	8
8	7 948	8 625	**0.322**	50 027	58 426	8 399	0 443	2 878	28 554	8
10†	10 136	10 750	0.307	80 691	90 763	10 072	0 355	1 785	34 240	8
10	10 020	10 750	**0.365**	78 855	90 763	11 908	0 355	1 826	40 483	8
12†	12 090	12 750	0.330	114 800	127 676	12 876	0 299	1 254	43 773	8
12	12 000	12 750	0.375	113 097	127 676	14 579	0 299	1 273	49 562	8

* Schedules 30 and 40 of the *American Standard for Wrought-Iron and Wrought-Steel Pipe, ASA* B36.10. Nominal thickness given in bold-faced type is identical with that for schedule 40.

† Thickness same as for schedule 30.

TABLE XX

Properties of Dichlorodifluoromethane, Freon 12

Sat Temp F	Pressure psia	Volume cu ft per lb		Enthalpy Btu per lb and Entropy above −40 F							
				Fnthalpy		Entropy		25 F Superheat		50 F Superheat	
		Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Enthalpy	Entropy	Enthalpy	Entropy
0	23 87	0 0110	1 637	8 25	78 21	0 01869	0 17091	81 71	0 17829	85 26	0 18547
5	26 51	0 0111	1 485	9 32	78 79	0 02097	0 17052	82 29	0 17786	85 89	0 18502
10	29 35	0 0112	1 351	10 39	79 36	0 02328	0 17015	82 90	0 17747	86 51	0 18460
15	32 44	0 0112	1 230	11 48	79 94	0 02556	0 16981	83 49	0 17710	87 13	0 18420
20	35 75	0 0113	1 121	12 55	80 49	0 02783	0 16949	84 09	0 17679	87 76	0 18382
25	39 33	0 0114	1 025	13 66	81 06	0 03008	0 16920	84 67	0 17643	88 37	0 18349
30	43 16	0 0115	0 939	14 76	81 61	0 03233	0 16887	85 25	0 17612	88 97	0 18315
35	47 28	0 0116	0 863	15 88	82 16	0 03458	0 16860	85 83	0 17582	89 56	0 18285
40	51 68	0 0116	0 792	17 00	82 71	0 03680	0 16833	86 41	0 17554	90 16	0 18256
45	56 38	0 0117	0 730	18 14	83 26	0 03904	0 16808	86 96	0 17528	90 76	0 18227
50	61 39	0 0118	0 673	19 27	83 78	0 04126	0 16785	87 54	0 17505	91 38	0 18203
55	66 74	0 0119	0 622	20 41	84 31	0 04348	0 16763	88 09	0 17482	91 93	0 18181
60	72 41	0 0119	0 575	21 57	84 82	0 04568	0 16741	88 64	0 17458	92 51	0 18155
65	78 44	0 0120	0 532	22 72	85 32	0 04789	0 16721	89 18	0 17436	93 11	0 18132
70	84 82	0 0121	0 493	23 90	85 82	0 05009	0 16701	89 72	0 17417	93 66	0 18114
75	91 60	0 0122	0 458	25 08	86 32	0 05229	0 16681	90 25	0 17397	94 23	0 18092
80	98 76	0 0123	0 425	26 28	86 80	0 05446	0 16662	90 78	0 17379	94 80	0 18075
85	106 40	0 0124	0 395	27 48	87 28	0 05665	0 16644	91 27	0 17361	95 33	0 18056
90	114 30	0 0125	0 368	28 70	87 74	0 05882	0 16624	91 77	0 17344	95 86	0 18040
95	122 80	0 0126	0 343	29 93	88 19	0 06100	0 16604	92 27	0 17323	96 39	0 18020
100	131 60	0 0127	0 319	31 16	88 62	0 06316	0 16584	92 75	0 17308	96 92	0 18004
105	140 90	0 0128	0 298	32 40	89 03	0 06534	0 16564	93 24	0 17291	97 46	0 17991
110	150 70	0 0129	0 277	33 65	89 43	0 06749	0 16542	93 66	0 17274	97 93	0 17976
115	161 00	0 0130	0 258	34 90	89 80	0 06965	0 16520	94 07	0 17253	98 40	0 17955
120	171 80	0 0132	0 240	36 16	90 15	0 07180	0 16495	94 47	0 17233	98 84	0 17939
125	183 10	0 0133	0 224	37 41	90 46	0 07395	0 16467	94 86	0 17210	99 26	0 17918
130	194 90	0 0134	0 208	38 69	90 76	0 07607	0 16438	95 25	0 17186	99 70	0 17897
135	207 40	0 0136	0 194	39 95	91 01	0 07816	0 16404	95 64	0 17161	100 13	0 17877
140	220 20	0 0138	0 180	41 24	91 24	0 08024	0 16363	96 03	0 17134	100 56	0 17856

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